PRODUCT ENGINEERING

VOLUME 7 NUMBER 4

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May Number

In the May issue will begin a series of articles on engineering department practice by J. K. Olsen, based on his experience, including that gained in the position of chief draftsman at Stewart-Warner.

An article of unusual interest will be one on the use of X-ray analysis in engineering design with specific reference to the X-ray examination of grain structure. Written by H. R. Isenburger, it will explain the principles used rather than attempting a profound scientific thesis.

"How to Specify Springs," by R. L. Stoughton of the Lee Spring Company, and "Selection of Material for Jewel Bearings" by B. W. St. Clair, president of the Mann Instrument Company, will also appear in May. The latter article, prepared especially for *Product Engineering*, summarizes the latest data of jewel bearing materials and gives ten typical designs of usual forms of such bearings.

Other articles in the May number include the concluding installment of the series on "Stress Calculations for Thin Aluminum Alloy Sections," which will relate to the calculation of members in torsion, and also another installment of the articles on alignment charts by M. G. Van Voorhis.

June Number

A special issue featuring electric and hydraulic drives and controls for machinery, including engineering data and typical designs, with complete explanation of the design and its functioning.

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Intimate Correspondence

Of Proved Value

PRODUCT ENGINEERING
NEW YORK

We have the sheets covering clutch design as taken from *Product Engineering*. The Johns-Manville Company also kindly sent us the reprint of a series of articles written by A. C. Rasmussen and which appeared in the 1931-1932 issues of *Product Engineering*. Several valuable suggestions were obtained from these articles and have been incorporated in the clutch we designed and built. Thanking you for your aid, we are, —A.C.K., *Chief Engineer*
Lancaster, Ohio

AND thus the published results of studies and research on the design of clutches for construction machinery developed by A. C. Rasmussen, chief engineer of Insley Manufacturing Company, find their way into the design of oil field equipment, designed by A.C.K.

All for an Ad

PRODUCT ENGINEERING
NEW YORK

Inasmuch as you are not able to send us a copy of your November issue, as requested in my letter of January 16, can you send us a tear sheet or a photostat of the advertisement run by the Dumore Company? Thank you for your cooperation. —R.C.J.
Boston, Mass.

WE are all out of copies of the November number. But anyone desiring tear sheets is certainly welcome to them, be they editorial pages or advertising.

Springs Wanted

PRODUCT ENGINEERING
NEW YORK

We are contemplating the use of Belleville Springs but are unable to find any manufacturers of such springs. Will you kindly advise us where such springs can be bought? —E.A.W.
Syracuse, N. Y.

WE have had a considerable number of inquiries on sources of supply for Belleville Springs, but up to the present time we have not been able to find any manufacturer who makes a specialty of them. If there are any such manufacturers, we like to know it.

Federal Bronze Address

PRODUCT ENGINEERING
NEW YORK

We would like to get in touch with the Federal Bronze Products Company, but are unable to locate their address. Their Mr. H. J. Shaffer had an article on page 457 of your December number. —C.J.L.
Columbus, Ohio

THIS is one of many letters we received asking the same question. The Federal Bronze Products Company is a relatively new company, hence the difficulty of finding their address which is 17-21 Newark Way, Maplewood, N. J.

Engineers' Licensing

PRODUCT ENGINEERING
NEW YORK

Like many others, I am not a college man but have studied a good deal in books and have taken correspondence courses. I have worked as tool designer, product designer, chief tool designer, plant foreman, and for the past fourteen years as supervisor in charge of a group of designers.

What steps should I take and just how far will this so-called "grandfather" clause go in securing for me the required license for professional engineer? —R.C.J.
Milwaukee, Wis.

MANY engineers in different states are facing this same question. Those desiring specific and authoritative information should write to the State Board of Education of their state.

Automatic Machinery

PRODUCT ENGINEERING
NEW YORK

Will you please send the writer the address of the Packer Machine Company whose polishing machine was described in January *Product Engineering*? I would also appreciate any data you may have on file as to where I can obtain a list of manufacturers who make automatic packaging machines. Do you publish a book on this type of machine? —D.D.D.
Tacoma, Washington

THE address of the Packer Machine Company is Meriden, Conn. The Packaging Machinery Manufacturers Institute, 342 Madison Ave., New York, N. Y., will be glad to furnish a list of their membership which includes most of the manufacturers of such equipment.

We know of no book dealing with the design of packaging machinery. Can any of our readers furnish it to us?

The Clipper's Complaint

PRODUCT ENGINEERING
NEW YORK

I was very much put out when I came to clip my March number of *Product Engineering* when I found that I could not trim the pages to 8½x11 without cutting off some of the diagrams. Pages 94, 95, 101 and 102 show what I mean. I wish you would avoid running the line drawings to the edge of the page. Is it really necessary? —R.M.S.
Cleveland, Ohio

WE will try to avoid this in the future and confine our "bleeds" to pictures that are not essential to an understanding of the text.

Gear Pump Formula Wanted

PRODUCT ENGINEERING
NEW YORK

We are interested in knowing whether or not there is any standard formula by which the displacement of a gear tooth in a gear type precision metering pump can be figured.

We would appreciate your advising us if there is any place where this information can be obtained. —J.P.L.
Philadelphia, Pa.

FRANKLY, we do not know of any such formula or the method of making such a calculation can be found. Perhaps some readers can enlighten us.

PRODUCT ENGINEERING
NEW YORK

I have recently finished reading the intimate correspondence column in the March issue of *Product Engineering*. One article by the editor causes me to write this letter to convey to you my most profound contempt for the stand taken on the question of unionism. May I ask what right you have in stating that a Mr. F.M.Z. probably did not have a union membership to hinder him in his progress? As a fair-minded man, I cannot tell you exactly what I think of your staff in making out of a wonderful magazine, a hodge-podge for some who think that they can create a hate for union people or principles. Why don't you stick to your *Product Engineering* and leave these problems to those who, at least, know the situation better than you.

My interest in the *Product Engineering* magazine has been through all of the years of its publication, and I sincerely hope that I will be able to continue to subscribe for it; but not if the magazine is to be turned into something that it was never intended for. —R.L.C.
San Francisco, Calif.

R. L. C. has given our terse statement an interpretation that was not in our mind when it was written. But upon rereading it we feel that he is well justified. We also agree that this particular subject has no logical place in *Product Engineering*.

PRODUCT ENGINEERING

GEORGE F. NORDENHOLT, *Editor*

MCGRAW-HILL PUBLISHING CO., INC. APRIL 1936 VOLUME SEVEN, NUMBER FOUR

Expensive Uncertainties

A LETTER from an official of the trade association embracing the steel founders complains about the difficulty of filling rush orders for a quality product. Makers of steel castings have a hard technical problem to solve in meeting some specifications and they feel that they should have sufficient time in which to study out the best method of making a particular casting. Under the pressure of competition they are reluctant to be too insistent. They are more likely to accept the order and do the best they can within the time limit set. As a consequence quality may be lower than it would be under more favorable circumstances.

This sort of handicap applies as much to many other parts and materials specified by the product engineer as it does to steel castings. Who is at fault and what should be done about it? An answer to this question must be found because rush orders are going to become more and more unpopular with suppliers as business picks up. By the same token suppliers are going to make fewer such concessions when their order books are full.

The two departments primarily concerned with parts and materials orders are generally the engineering and the purchasing departments. Each of them should probably take some part of the blame but the chances are that the purchasing department is the lesser offender. Purchasing, after all, is more or less of a routine job after the engineering department has specified its needs, unless the purchasing agent takes too long in playing one supplier against another to get the best possible price.

Engineering and design being classified in the field of professional rather than commercial activity, it is far easier for the engineer to take his time about making up his mind on just what he wants and presenting the usual alibis for his procrastination. Every production man and almost every purchasing agent can cite case after case where engineering delays have put him in a hole. Hence the endless arguments between these department chiefs and the engineers.

This editorial is a plea to product engineers to adopt a bit more of the practical attitude toward the specifying function, and to push their professional prerogatives into the background. The other fellow likes to have time enough to do his part of the job as well as he knows how, even to be a bit professional about it. If the engineer will give him that chance the overall result will be a better product and more credit for the designer of the product. Unavoidable delays in setting specifications cause enough trouble in busy times. Why not take steps to cut out the avoidable ones? Such uncertainties are expensive.

Development and

U. A. WHITAKER
Director of Development and Design
The Hoover Company

Fig. 1—Entrance to the home of Hoover engineering at North Canton, Ohio

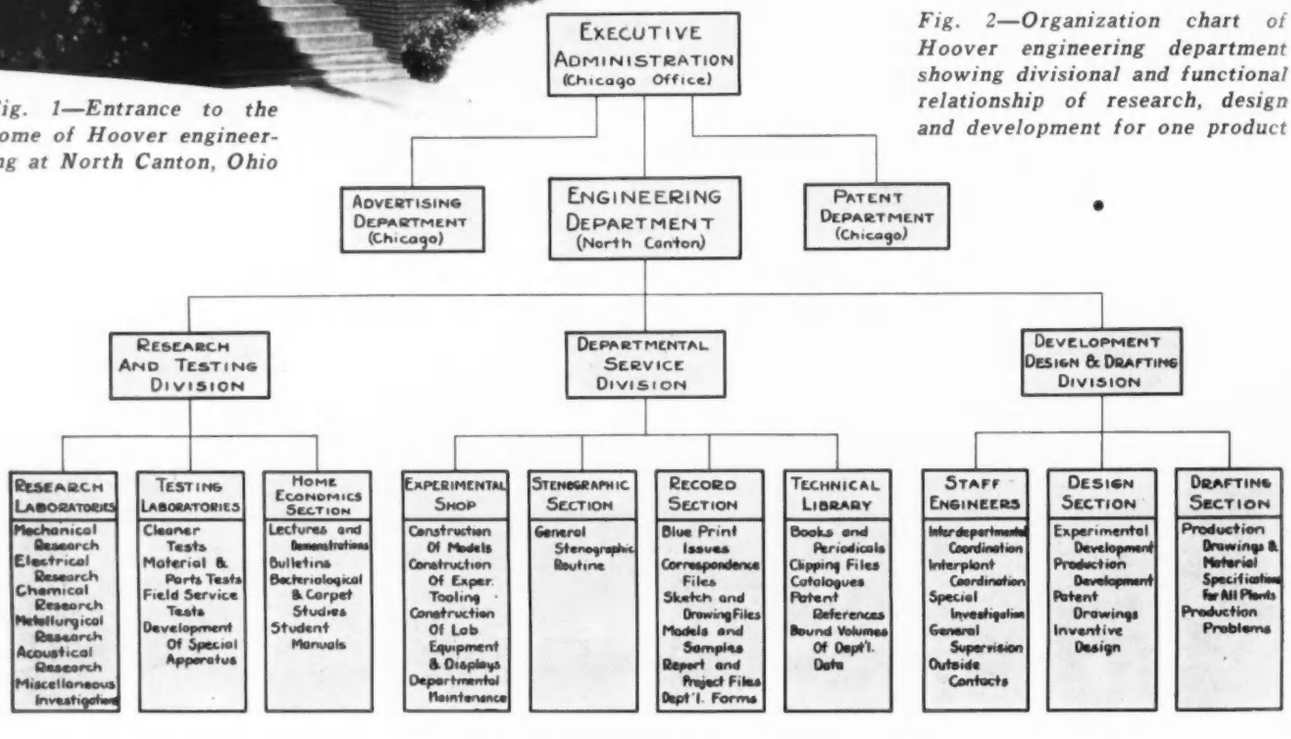


Fig. 2—Organization chart of Hoover engineering department showing divisional and functional relationship of research, design and development for one product

IT MAY be difficult to understand how 130 men can be profitably employed in the engineering department of The Hoover Company, which makes only one product and that of such apparent simplicity as an electric vacuum cleaner. The explanation lies in the unusual setup of the concern.

The management, directors, and stockholders have almost exclusively only the one business interest, this being the manufacture of electric vacuum cleaners. In view of the non-diversity of its product the company has considered as wise insurance the maintenance of a large enough engineering department to place it in a predominant position in relation to the rest of the industry as to technical knowledge, patents and other phases. This insurance offsets, to a large degree, the otherwise hazardous condition of depending entirely on one product for the success of the company. This large engineering staff is maintained in a modern building of four stories having 60,000 sq.ft. floor space, adequately equipped with modern apparatus and machinery.

In general, the engineering department is functionally divided into three major divisions: one being the design, development and drafting section. This division makes all the contacts with other divisions of the company and with outside sources. The second division is the research and testing laboratories, including all research developments, electrical motor design, fan and fan chamber design and testing necessarily carried out in a laboratory. The third is the service division which includes stenographic, record and blueprint departments, library and experimental construction shop. Approximately half the work of the engineering department is related to engineering services which have to do with the routine maintenance of the quality of the product. The remaining half is applied toward the active development and preparation of new models for production. It is the development of new models that will be discussed in this article.

Inception of New Models

New models have their inception, years before the actual development starts, in the fundamental research carried on in the various departments of the research laboratories, and in the inventive activities of the development and design division. Ordinarily the program for new models is kept outlined about five years in advance of production, the engineering department generally working on four or more new cleaner designs at one time. The actual formation of the preliminary plan and proposal for a new model is generally initiated in

Engineering Procedure

When the continued success of a business depends wholly on one product, the importance of design overshadows all other factors. Here is told how the engineering department, with a staff of 130, keeps the Hoover vacuum sweeper in a position of leadership by the continuous development of advanced designs

the development and design division. A rough specification is prepared and the suggested new model is then proposed to the management through the vice-president in charge of engineering. These preliminary proposals for new models are usually several years ahead of production, although not always so.

If the new proposal is accepted it is given a suitable position on the engineering department work program according to the urgency of completing the engineering work. If the particular model is desired at a certain date a schedule is set up for all the departments.

Preliminary Development

After the proposed new model has been accepted by the management the job is assigned to a staff engineer. He is responsible for the necessary coordination between the patent, sales, purchasing, production, advertising, and art departments. This coordinator, or staff engineer, is one of a group of men who have been selected because they are familiar with all problems.

When completed, the staff engineer sends the drawing to the cost department and cooperates with it to compile an official cost estimate. Mechanisms that seem likely to raise patent questions are taken up with that department at once. Considerable laboratory work also may be started at this time to discover design weaknesses. In this way, any mechanisms that do not function properly or are out of line either from the patent or the cost viewpoint are changed at once, thus avoiding more costly changes later.

After completion of the preliminary work, the job is turned over to the design room, where it is divided for greatest convenience and assigned to various engineers all under the supervision of the chief designer. Necessary contact between these designers and other sections of the engineering department, or with outside sources, is made through the staff engineer, who procures all information and the answers to any questions that may arise. As the design progresses, preliminary models of various mechanism and parts that are new to the particular model are made. Tests are started in laboratories and field to obtain early information on such parts before more complete tests on the later models are made.

Working Model

As soon as layouts covering the major portion of the machine are finished, a complete working model and possibly several laboratory models are built from the design layouts. These models are necessarily crude as the design layouts are not completely dimensioned. However, the laboratory models serve for a large part of the tests that must be made and the complete working model is made sufficiently correct to serve for an appearance study.

Attention is given to the appearance, both of the machine as a whole and of its individual parts. To assist in the appearance details, the services of an outstanding industrial designer are retained. All parts having to do with the final appearance are submitted for

Fig. 3—A staff engineer acts as coordinator between patent, sales, purchasing, production, advertising, and art departments

his criticisms or suggestions and the entire general appearance of the machine is then worked out from his proposals.

Upon its completion the first functional model is submitted to the management. At this time the general functional characteristics of the machine have already been established as have been the proposed production schedule and the estimated production cost. After the model has been presented for approval it is usually necessary to make certain changes to conform with the policy the management wishes to follow in its merchandising plans. The sales department is also contacted for its reaction and to uncover any possible weakness from the viewpoint of merchandising and demonstrating. It is also necessary to determine any demonstration equipment or special devices needed by the salesmen or demonstrators in the various stores.

The necessary revisions recommended as a result of these several conferences are made and the layouts are then turned over to the drafting room to prepare the necessary production drawings. When these drawings are completed, the cost department gives a more refined estimate, the patent department gives final approval, and the experimental shop build models accordingly.

From the time production drawings are started, close cooperation is maintained between the engineering department and the production department and with outside suppliers. Some of the parts are actually developed to a production basis in the engineering department before being turned over to the production department.

Final drawings are also given to the purchasing department as soon as they are completed so that preliminary quotations on such parts as must be purchased from outside suppliers can be obtained as soon as possible. At this time the engineering department undertakes the development of an approved source of supply on such parts that require high quality workmanship and careful control. The engineering department requires possible suppliers to submit samples, and necessary tests are made and specifications established. An effort is made to obtain at least two approved sources on each part so that the purchasing department has some discretion when placing orders.

Control test procedure is set up on all important parts or material and the sources of supply are informed about the tests to which their material will be subjected. An extended period of tests and changes are often required before the materials are finally improved to the point where they meet the specifications and are acceptable. By this system much trouble in the field is avoided.

Testing of Final Models

In addition to all the preliminary or partial models, ten complete machines are usually built in the experimental shop in exact accordance with the detailed drawings. Field tests are made on these machines for general efficiency, noise, durability, and other qualities. All the tests to which the machines and their component parts are subjected are too numerous to be outlined in this article. Some of the more important are cleaning effectiveness under laboratory and home conditions; numerous abuse and breakdown tests; motor efficiency, temperature, and lift tests; belt, bearings, and carbon brush life; aging qualities and resistance of all rubber

Fig. 6 (Left) — Here test models are given accelerated service tests equivalent to 10 or 15 years of home use and break-down tests are conducted on all accessories

Fig. 7 (Right) — The sound room has 9-in. concrete walls and is mounted on coiled springs, isolating it from the rest of the building

Fig. 4 (Left)—Tests to determine motor characteristics such as temperature rise, power consumption and carbon brush life are made in a separate electrical engineering laboratory

Fig. 5 (Right)—After the preliminary and partial models, ten complete cleaners are built exactly to specifications in the experimental shop

parts to light; twisting, abrasion, and flexing tests on the electric cord; and the durability and electrical properties of insulation and switches.

Abuse tests to which the machine and its component parts are subjected, have been established over a number of years and are designed to represent actual service conditions. The reliability of these accelerated abuse tests are then checked by a fleet of home service cleaners and those maintained in various commercial establishments where they are subjected to severe service. These test models are given as much use in a year as an average cleaner gets in ten to fifteen years in a home. Many of the home service cleaners and all the cleaners placed in commercial establishments are equipped with electrical timing devices to indicate the number of hours the machine has been in use.

The cleaning effectiveness tests are conducted under exacting control. Relatively, these are the most extensive of any of the tests made. The company states in its advertising literature that its product collects more dirt per minute than any other cleaner. Therefore, the company goes to great expense to conduct such tests accurately and conclusively to the end that the product will meet the advertised claims of efficiency. Laboratory tests are made on company-owned rugs placed in various stores, theatres, and other public buildings in the community. Comparative tests under actual cleaning conditions are also made in private homes in the district. A total of about 30 runs are made to establish the cleaning effectiveness of any particular new model.

All motors are developed in the electrical laboratory and this department is held responsible for carrying out the necessary tests to determine motor characteristics, such as temperature rise, carbon brush, life, power consumption, and others.

As noise is one of the most important factors to be contended with, a completely equipped sound research laboratory is maintained. A specially built sound room, isolated from the rest of the building, is submerged below the basement floor. This room is made of concrete with walls 9 in. thick. It weighs approximately 50 tons, and is mounted on coil springs. Here the motors, fan, and fan chamber combinations, and agitators are tested separately, necessary alterations being made to obtain the quietest arrangement. These elements are then tested, combined in the machine as it is to be built, and, if necessary, further modifications and refinements are made to get as quiet a machine as possible.

One of the important functions of the engineering department arises from the service problems created by the new model. One staff engineer gives his entire time to studying service problems and to transmitting the necessary information to the service departments over the country. He also sees that proper consideration is given to the service problems in the design of the machine.

Another factor that must be taken into consideration is the relation of the new design to foreign activities. Plants are maintained in Hamilton (Ontario), London, and Paris. A decision must be reached as to when the new model will be introduced into such countries, and, if so, what parts will be supplied from this country. The questions that arise in connection with foreign plants can only be appreciated by those who have had experience with such plants.

PRODUCT DEVELOPMENTS

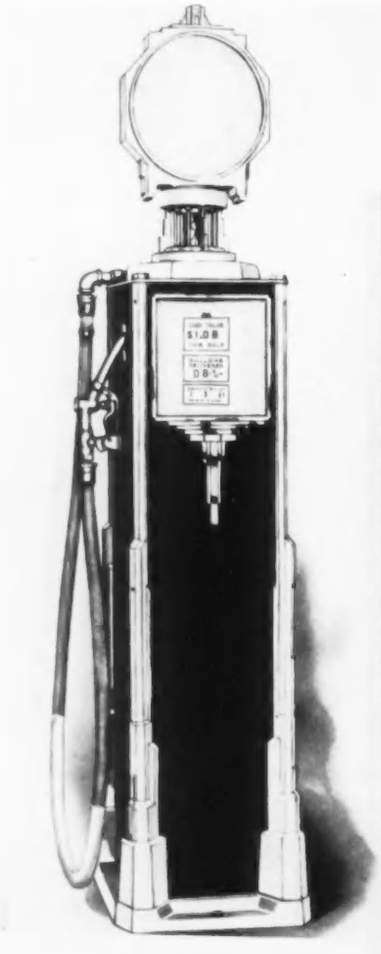
• *Modernized Gas Dispensing Pumps*

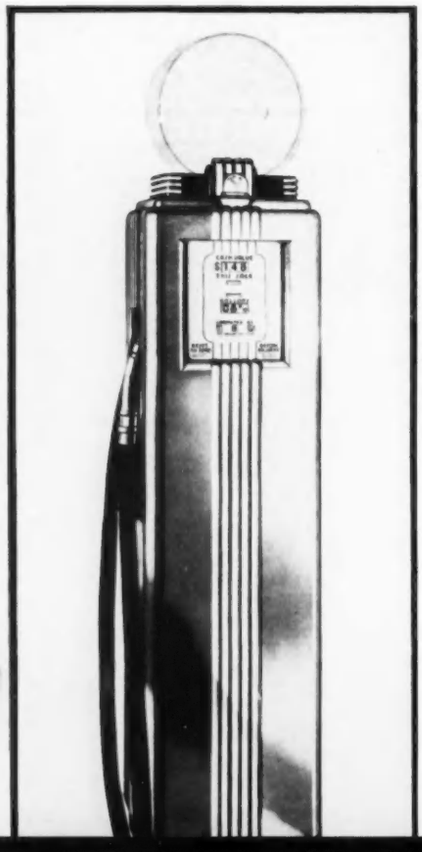
Style changes during the past two or three years are sharply contrasted in the old and new gasoline computing pumps made by Erie Meter Systems, Inc. As shown in the illustration, decoration extensively used on the old model at the left has been stripped off. Surfaces have been smoothed and corners rounded, giving the new pump a more dignified severity.

But in addition to bringing the appearance up to date, this redesign by Wilbur Henry Adams was responsibile for important reductions in the cost of the case. Only one die is needed for both the front and back. A standard round globe is used and the height is reduced.

Although the inside of the pump is essentially the same, the reduction in the cost of the case made possible a stronger structure and a number of refinements in the pump mechanism without increasing the total cost of the pump. A heavier channel iron substructure is used. Units are arranged more compactly, bringing the dial down low.

Last year's model gasoline computing pump (below) with the new model shown next to it





• *Truck Bodies of Stainless Steel*

Bakery products are being delivered hot from the ovens to the 25 stores of the Hanscom Baking Corporation in New York City, in lightweight, stainless steel, insulated truck bodies designed and built by the Edward G. Budd Manufacturing Company. Both the structural framing and sheathing are of stainless steel. With the light weight thus attained, load capacity was increased to approximately five tons.

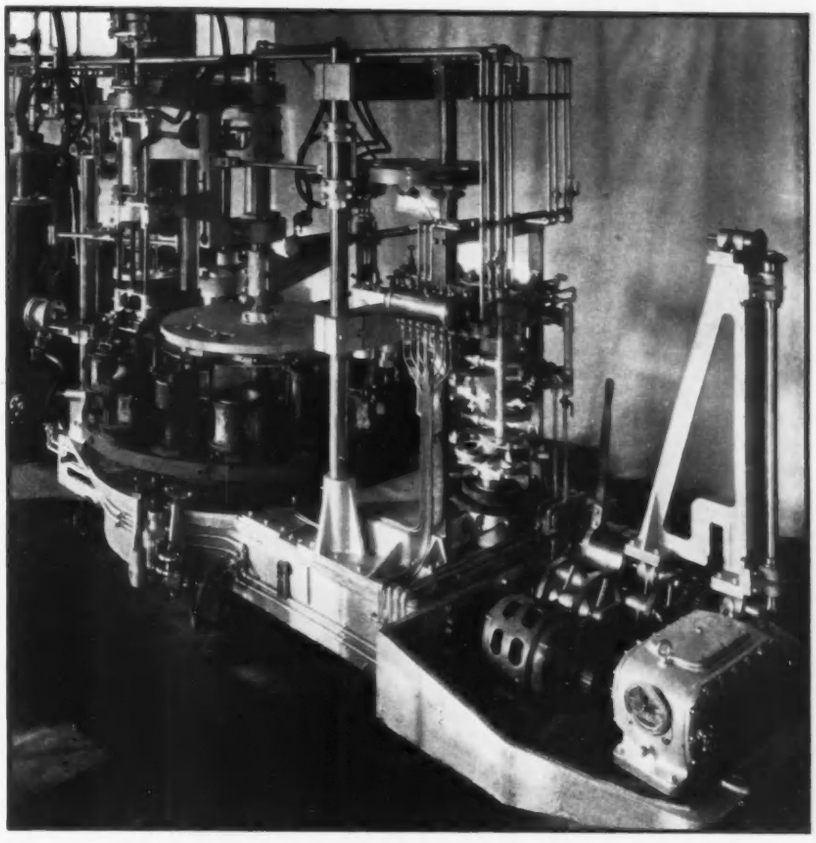
Insulated bakery truck body built by Budd with stainless steel framing and sheathing. Capacity is approximately five tons and the bakery products are kept hot for delivery

As can be seen in the illustration, the exterior of the body is corrugated stainless steel. Color contrast is obtained by painting some of the corrugations green to match the color of the cabin, and formed letters of stainless steel are attached to the corrugated surface. Interior trim is plywood and the floor is oak bolted to the floor beams. Insulation in the walls keeps the product hot until it is delivered to the retail stores.

Pressure Circulation for Automobile Engines

It is not often that engine or car design is altered to increase the efficiency of an accessory, but in one current design, modifications made to improve performance of automobile heaters also resulted in more efficient engine cooling as well. In this redesign, the conventional suction water pump, shown in Fig. 1, was replaced by the pressure system shown in Fig. 2. Thus, the water is pushed into the bottom of the cylinder block instead of being pulled from the top.

This change not only gave a more positive flow of hot water to the heater system but also resulted in more efficient cooling. The conventional suction type water pump reduces the pressure at the top of the cylinder block, thereby lowering the boiling point of the water in the cylinder head. Steam pockets form and cause irregular flow when the water system is not completely filled. With the pressure pump system increased pressure is created in the head, thereby raising the boiling point and tending toward the elimination of steam pockets.

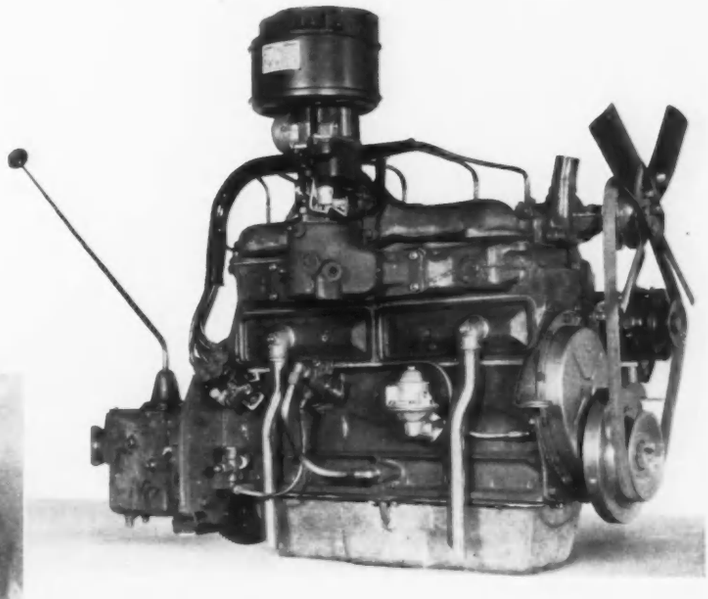


By means of a Link-Belt variable-speed transmission, the Miller bottle blowing machine can be operated at the highest practical speed

Adjustable Speed for Bottle Blowing Machine

In the accompanying picture is shown one of the latest machines built by Wm. J. Miller, Inc., for producing pressed glass tableware and novelties, wide-mouthed bottles and jars, lantern globes, thin-blown tumblers, lamp-shades and similar shapes. This machine mounts 8 blank and 8 blow moulds and is designed for operation with an automatic feeder which feeds the molten glass from the furnace to the machine.

In such a machine, the maximum speed practical for the production of good ware is limited according to



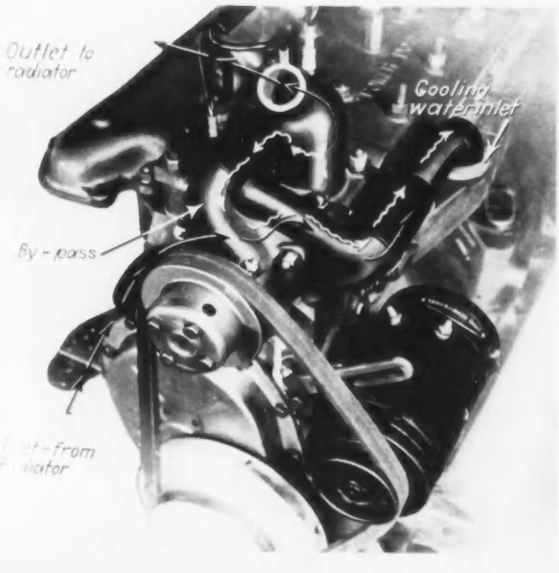


Fig. 1 (Above)—Conventional water pump installation

Fig. 2 (Left)—Redesigned pump installation forces water into cylinder block under positive pressure, resulting in improved cooling

the temperature of the glass being blown and the shape and thickness of the article being made. To be able to always operate the machine at the highest practical speed, some form of variable speed control is necessary in order that the speed can be adjusted to the highest speed that the particular conditions permit. Change gears would not be practical as they would require stopping the machine in order to make the desired speed change, thereby entailing a considerable production loss. Even then the best operating speed might not be obtainable.

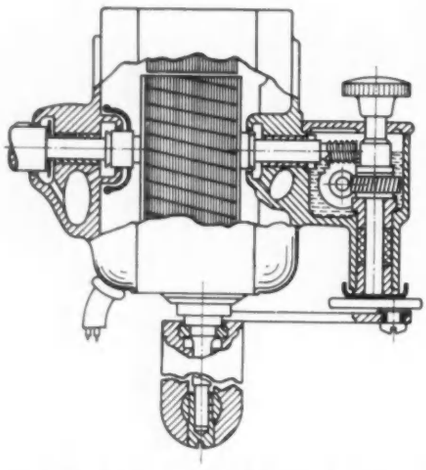
In the machine illustrated here a Link-Belt variable-speed transmission is used, as shown in the lower right-hand corner of the illustration. This permits continuous adjustment of speed to give a rate of production in the range of 15 to 36 pieces per min., without requiring either slowing down or stopping of the machine.

• *Lubricant Seals for Fan Motors*

Oil leakage, one of the most objectionable weaknesses heretofore found in fans, is prevented in Wagner fans by a completely revised system of lubrication. As shown in the accompanying diagram, slingers machined on the thrust collars prevent the passage of oil into the motor from either end. Centrifugal force throws the oil outward against the walls of the collection chambers, from where the oil drains back through holes to the wells. A disk acts as a slinger at the front end of the fan, throwing any escaping oil outwards against the wall of the collection chamber.

To prevent the suction created by the fan blades from drawing oil out of the well and past the front disk, a bell-shaped draft seal is provided at the suction end inside the motor. Within this cup is set up an air whirl sufficient to prevent all air suction through the well.

Grease leakage in the oscillating mechanism is prevented by a series of felt packing washers. A gasket prevents leakage of grease below the lid, and the felt washers fitting closely, a close fit around the vertical oscillator shaft are safeguards against escape of grease at that point.



Seals and slingers prevent loss of oil and grease in the new Wagner fans

• *Changes in Combustion Chambers*

As a result of the recent increases in compression ratios, American automobile engineering practice is veering away from the Ricardo type combustion chamber with its venturi form. At the time the Ricardo design was developed compression ratios were about 4½ or 5 to 1, and high engine speed could not be obtained because gas mixtures could not be made to burn fast enough. Turbulence was necessary to speed up the burning process. With the advent of modern high compression engines of 6 and 7 to 1 ratio, the use of a restriction or other means of increasing turbulence is unnecessary.

With high compression it is more important to control the rate of flame front advance than to introduce mechanical turbulence. Hence, chambers must be designed for the best combustion control consistent with good filling characteristics, and volumetric efficiency must be maintained.

Recent laboratory tests have shown that a restriction in the cylinder is to be avoided if possible, rather than to be placed there deliberately. These results indicate that the throat area should be based on the swept volume of the cylinder rather than from the valve area, as taught by Ricardo. The smallest area throughout the induction system, excluding the carburetor venturi, therefore should be at the valves.

• *Random Jottings About New Developments*

In a new series of electric excavators built by Harnischfeger every motion in hoisting, crowding, swinging and traveling is powered by an independent motor separately controlled by a push button from the operator's station. Brakes and clutches are hydraulically controlled, eliminating hand levers, pedals, reach rods and bell cranks.

* * *

Flexible blades of tough, molded rubber, used on new Samson electric fans, although rigid enough to deliver a steady current of air, are sufficiently soft so that no guard is necessary.

* * *

A new type of aluminum piston developed by Bohn Aluminum & Brass Corporation, under comparative tests against standard aluminum pistons shows increased horsepower and decreased friction as a result of closer and more accurate fit permissible in the cylinder bore.

Carriages and frame assemblies for a new comparator microscope built by Stupakoff Laboratories are constructed of aluminum alloy for light weight. All other moving parts are of non-corrosive metals, and illumination is built in.

* * *

For mowing machine gears, the Massey-Harris Company, Ltd., Canadian builders of agricultural machinery, are using an iron containing 2.75 per cent carbon, 1.3 per cent nickel, and 0.5 per cent molybdenum in order to obtain the required high strength and wear resistance.

* * *

On the new Ediphone, the dictator mouthpiece is of Durez not only for better accoustic qualities, but also because it is light in weight and the finish is permanent.

* * *

Cylinder heads and barrels on Wright Cyclone air-cooled aircraft engines are Bonderized to provide a rust-proof surface that resists corrosion from salt water.

Stress Calculations for Thin Aluminum Alloy Sections

S. A. KILPATRICK, *Test Engineer* and O. J. SCHAEFER, *Stress Analyst*
The Glenn L. Martin Company

PART III—Hollow Girders

An analysis of box beams or hollow girders with examples showing how to calculate them. For the derivation of the equations for unsymmetrical sections for semi-monocoque structures the authors are indebted to Guy L. Bryan, Jr., who is in charge of the structural department of The Glenn L. Martin Company. He also gave valuable suggestions and constructive criticisms on these articles.

THE box beam or hollow girder may be designed as a monocoque or semi-monocoque structure. A pure monocoque structure, strictly defined, consists of a thin hollow shell without transverse or longitudinal stiffeners. This type of structure is seldom encountered in practice because it has a low strength to weight ratio and deforms appreciably under load. If the shell is reinforced with longitudinal stiffeners or transverse frames, or both, it is classified as a semi-monocoque structure. Such structures utilizing both longitudinal and transverse stiffeners are used extensively.

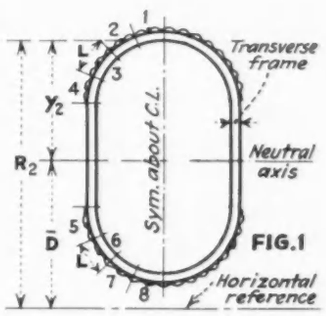


Fig. 1—Showing how the maximum bending stresses in a structure consisting of corrugated sheet chord sections and thin web side skin can be closely approximated by the method explained in the text

Let us first consider a structure as shown in Fig. 1, consisting of corrugated sheet chord sections and thin web side skin. In such a structure, the maximum bending stresses can be closely approximated in accordance with the theory of elastic bending, by the formula:

$$f_b = \frac{My}{I_x} \qquad (1)$$

In this equation M is the applied bending moment, y the distance from the neutral axis to the fibers in question, and I_x is the moment of inertia of the cross-section.

In determining the neutral axis and the moment of inertia of the section shown in Fig. 1, the side skin is neglected because the portion in compression is incapable of taking any appreciable compression stresses. This is made evident in tests by the formation of dimples in the side sheet. These dimples should cause no concern since they indicate merely that the skin has failed elastically. After removal of the applied load these dimples disappear. The portion of the side skin in tension is neglected in order to simplify the computations. The error resulting from this simplification is negligible.

The usual method of determining the neutral axis and the moment of inertia for a section as shown in Fig. 1, is as follows: first, divide the corrugated sheet chord sections into a convenient number of units of length L. The lengths should be chosen small enough so that the moment of inertia of each unit about its own axis is small compared to the total moment of inertia. The centroids of these units are indicated by small circles, numbered *1, 2, 3,* . . . etc., as shown in Fig. 1. If we represent the areas of these units by A_1, A_2, A_3, . . . etc , and the distance of each unit from the reference line by R_1, R_2, R_3, . . . etc., the distance $\bar{D}$ between the neutral axis of the entire section and the arbitrarily chosen reference line can be computed from:

$$\bar{D} = \frac{A_1R_1 + A_2R_2 + A_3R_3 + \ldots . A_sR_s}{A_1 + A_2 + A_3 + \ldots . A_s} \qquad (2)$$

The moment of inertia of the section is simply the summation of each area times the square of its distance

Table I—Computation of Neutral Axis and Moment of Inertia

Unit	Area	Dist. from Centroids to Ref. Line	Area Moment about Ref. Line	Dist. from Centroids to Neutral Axis	Area Times y^2
1	A_1	R_1	A_1R_1	y_1	$A_1y_1^2$
2	A_2	R_2	A_2R_2	y_2	$A_2y_2^2$
3	A_3	R_3	A_3R_3	y_3	$A_3y_3^2$
.	.	.	. .	.	. .
.	.	.	. .	.	. .
.	A_s	R_s	A_sR_s	y_s	$A_sy_s^2$
	ΣA		ΣAR		ΣAy^2

$\bar{D} = \frac{\Sigma AR}{\Sigma A}$ Moment of Inertia $= 2\,\Sigma Ay^2$

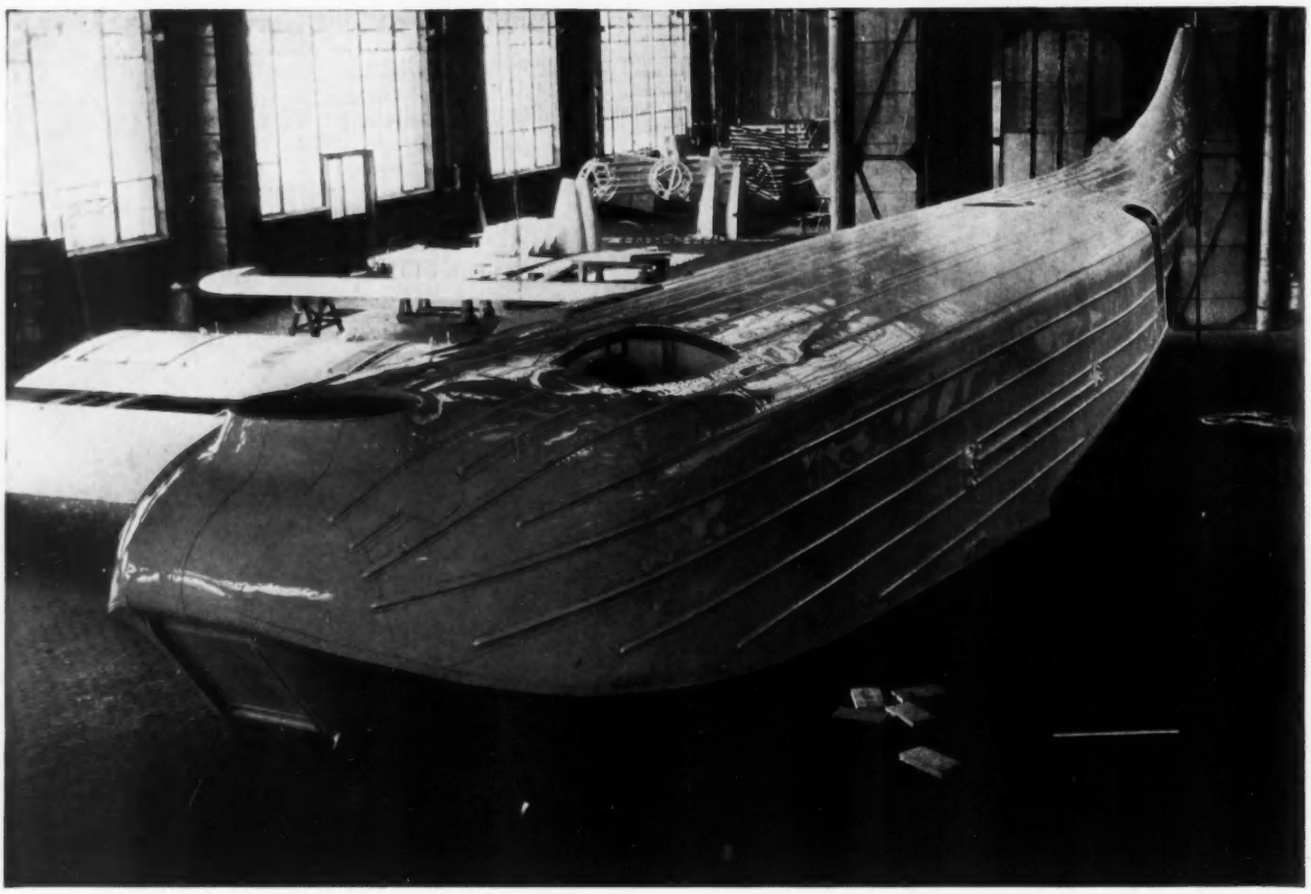


Fig. 2—A semi-monocoque hull with smooth sheet covering and longitudinal stringers on top and sides placed on the outside so that transverse frames need not be cut, thereby simplifying construction

Table II—Calculation of the Properties of the Section Shown in Fig. 3

(1) Unit	(2) Area	(3) R	(4) A R	(5) r	(6) A r	(7) x	(8) A x^2	(9) *y	(10) A y^2	(11) Axy
b	0.864	28.37	24.51	0	0	—53.43	2,466	21.56	402	— 995
1	0.938	29.16	27.35	5.47	5.13	—47.96	2,158	22.35	468	1,006
2	0.938	30.55	28.66	16.40	15.38	—37.03	1,286	23.74	529	— 824
3	0.938	31.36	29.42	27.34	25.64	—26.09	638	24.55	565	— 601
4	0.938	31.63	29.67	38.27	35.90	—15.16	216	24.82	578	— 353
5	0.938	31.59	29.63	49.21	46.16	— 4.22	17	24.78	576	— 98
6	0.938	31.19	29.26	60.14	56.41	6.71	42	24.38	558	153
..	..	..	..	..	..	..	..	..	..	..
..	..	..	..	..	..	..	..	..	..	..
14	0.350	—21.32	— 7.46	38.27	13.39	—15.16	80	—28.13	277	149
15	0.350	—21.46	— 7.51	49.21	17.22	— 4.22	6	—28.27	280	42
16	0.350	—21.29	— 7.45	60.14	21.05	6.71	16	—28.10	276	— 66
17	0.350	—20.82	— 7.29	71.08	24.88	17.65	109	—27.63	267	— 171
18	0.350	—20.20	— 7.07	82.01	28.70	28.58	286	—27.01	255	— 270
19	0.350	—19.45	— 6.81	92.95	32.53	39.52	547	—26.26	241	— 363
20	0.350	—18.58	— 6.50	103.88	36.36	50.45	891	—25.39	226	— 448
e	3.000	—17.45	— 52.35	109.35	328.05	55.92	9,381	—24.26	1,766	— 4,070
Totals	19.982		136.02		1,067.61		34,727		11,766	— 727

**Measured from center of corrugations.*

C. G. of Section $= \frac{136.02}{19.982} = 6.81$ in. above Reference line

C. G. of Section $= \frac{1,067.61}{19.982} = 53.43$ in. to the right of Reference line

$I_x = 11{,}766$ in.^{4} $I_y = 34{,}727$ in.^{4} $H = -727$ in.^{4}

Table III—Tabulation of Stress Computations

Points	x	*y	85x	—1086y	Stress	Points	x	*y	85x	—1086y	Stress
b	53.43	22.16	—4,540	—24,070	—28,610	d	—53.43	—25.24	—4,540	27,410	22,870
1	47.96	23.10	—4,080	—25,090	—29,170	11	—47.96	—25.96	—4,080	28,190	24,110
2	37.03	24.49	—3,150	—26,600	—29,750	12	—37.03	—26.97	—3,150	29,290	26,140
3	26.09	25.30	—2,220	—27,480	—29,700	13	—26.09	—27.70	—2,220	30,080	27,860
4	15.16	25.57	—1,290	—27,770	—29,060	14	—15.16	—28.13	—1,290	30,550	29,260
5	4.22	25.53	— 360	—27,730	—28,090	15	— 4.22	—28.27	— 360	30,700	30,340
6	6.71	25.13	570	—27,290	—26,720	16	6.71	—28.10	570	30,520	31,090
7	17.65	24.61	1,500	—26,730	—25,230	e	55.92	—24.86	4,750	27,000	31,750
Splice	23.11	24.19	1,960	—26,270	—24,310						

**Measured from extreme fibers of corrugations.*

to the neutral axis, which, expressed in equation form, is as follows:

$$I_x = 2(A_1y_1^2 + A_2y_2^2 + A_3y_3^2 + \ldots . A_8y_8^2) \qquad (3)$$

The above computations are best done by tabulations as given in Table I.

This method is quite satisfactory for a case in which the section is symmetrical and the plane of the applied bending moment is normal to the neutral axis. Very often the designer of semi-monocoque structures is confronted with the design of an unsymmetrical section. The use of the simple beam formula $f_b = My/I_x$ for any unsymmetrical section or loading condition may lead to serious errors. This subject has been treated in foreign publications, but little has been presented in English texts and will therefore be presented in detail in this article.

Let us take the structure shown in Fig. 3 as an example of an unsymmetrical box beam. The fiber stress at any point on the beam can be expressed by the equation:

$$f_b = \frac{(M_yH - M_xI_y)y + (M_xH - M_yI_x)x}{I_xI_y - H^2} \qquad (4)$$

For this equation the reference lines $X - X$ and $Y - Y$ are any convenient set of rectangular axes passing through the centroid of the cross-section.

M_x = component of moment perpendicular to $X - X$ axis
M_y = component of moment perpendicular to $Y - Y$ axis
I_x = Moment of inertia of section about $X - X$ axis
I_y = Moment of inertia of section about $Y - Y$ axis
H = product of inertia about $X - X$ and $Y - Y$ axes. This factor is equal to the sum of the products of each area multiplied by both of its coordinates.
x and y = distances between the centroids of areas and the $Y - Y$ and $X - X$ axes, respectively.

The above equation is a general expression for the normal stress at any point on the structure. The plane of the applied bending moment producing this stress may be acting at any angle to the $X - X$ axis. Before substituting in the equation, the applied bending moment must be resolved into components about the $X - X$ and $Y - Y$ axes.

From this general expression one may easily derive the usually encountered case of bending. For example, if the $X - X$ and $Y - Y$ axes are principal axes, H is equal to zero and the expression becomes:

$$f_b = \frac{M_xy}{I_x} + \frac{M_yx}{I_y}$$

For the extremely limited cases when H is equal to zero and the angle between the plane of the applied bending moment and the $X - X$ axis is equal to 90 deg., Equation (4) becomes $f_b = My/I_x$, which is in agreement with Equation (1). This formula is valid only if the cross-section is symmetrical about at least one axis and the plane of the resulting bending moment contains this reference axis.

In Table II, which is self-explanatory together with Fig. 3, all necessary calculations leading up to the computations of beam stresses are shown.

The following convention of signs must be adhered to in order that the sign of the resultant stress, if negative, will denote compression, and if positive, tension:

(1) Distances above the $X - X$ axis are positive.
(2) Distances to the right of the $Y - Y$ axis are positive.
(3) Moments about the $X - X$ axis, producing compression in the fibers above the $X - X$ axis, are positive.
(4) Moments about the $Y - Y$ axis producing tension in the fibers to the right of the $Y - Y$ axis are positive.

The stresses in the fibers will now be computed, assuming that $M_x = +\ 12{,}840{,}000$ in. lb. and $M_y = -\ 3{,}724{,}000$ in. lb. Substituting these moments and the values computed in Table II, in Equation (4):

$$f_b = \frac{[-\ 3{,}724{,}000(-\ 727) - 12{,}840{,}000(34{,}727)]y}{11{,}766 \times 34{,}727 - 727^2} + \frac{[12{,}840{,}000(-\ 727) - (-\ 3{,}724{,}000 \times 11{,}766)]x}{11{,}766 \times 34{,}727 - 727^2}$$

$$= -\ 1.086y + 85x$$

Thus far, nothing has been said about allowable stresses to be used for chord sections. If these chord sections consist of corrugated sheet, as used in the preceding examples, the problem is quite simple. (For information regarding allowable stresses of corrugations see Feb. *P.E.*, pages 50, 51, and 80). The column length of corrugations used in semi-monocoque construction is taken equal to the distance between transverse frames, and the coefficient of end restraint is assumed equal to one, in the usual design. If the cor-

Fig. 3—An example of an unsymmetrical box beam, the analysis of which is given in the text. Fig. 4—Allowable sheet load is a function of coefficient C as given by these curves

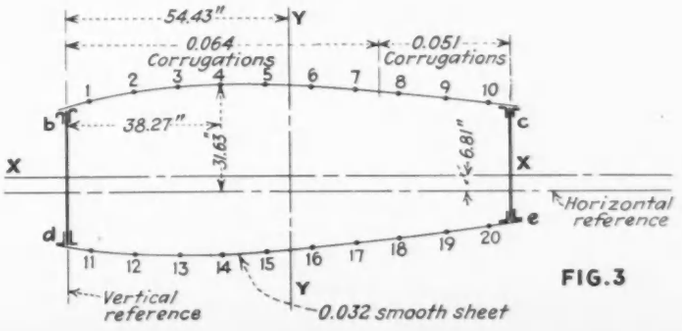


FIG. 3

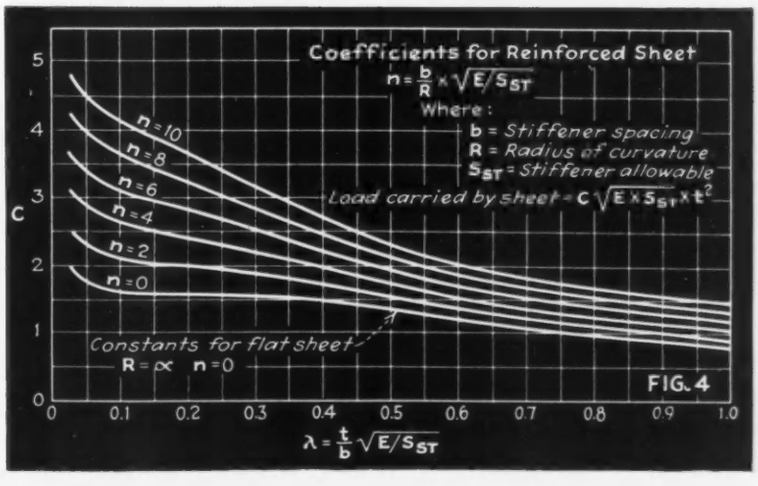


FIG. 4

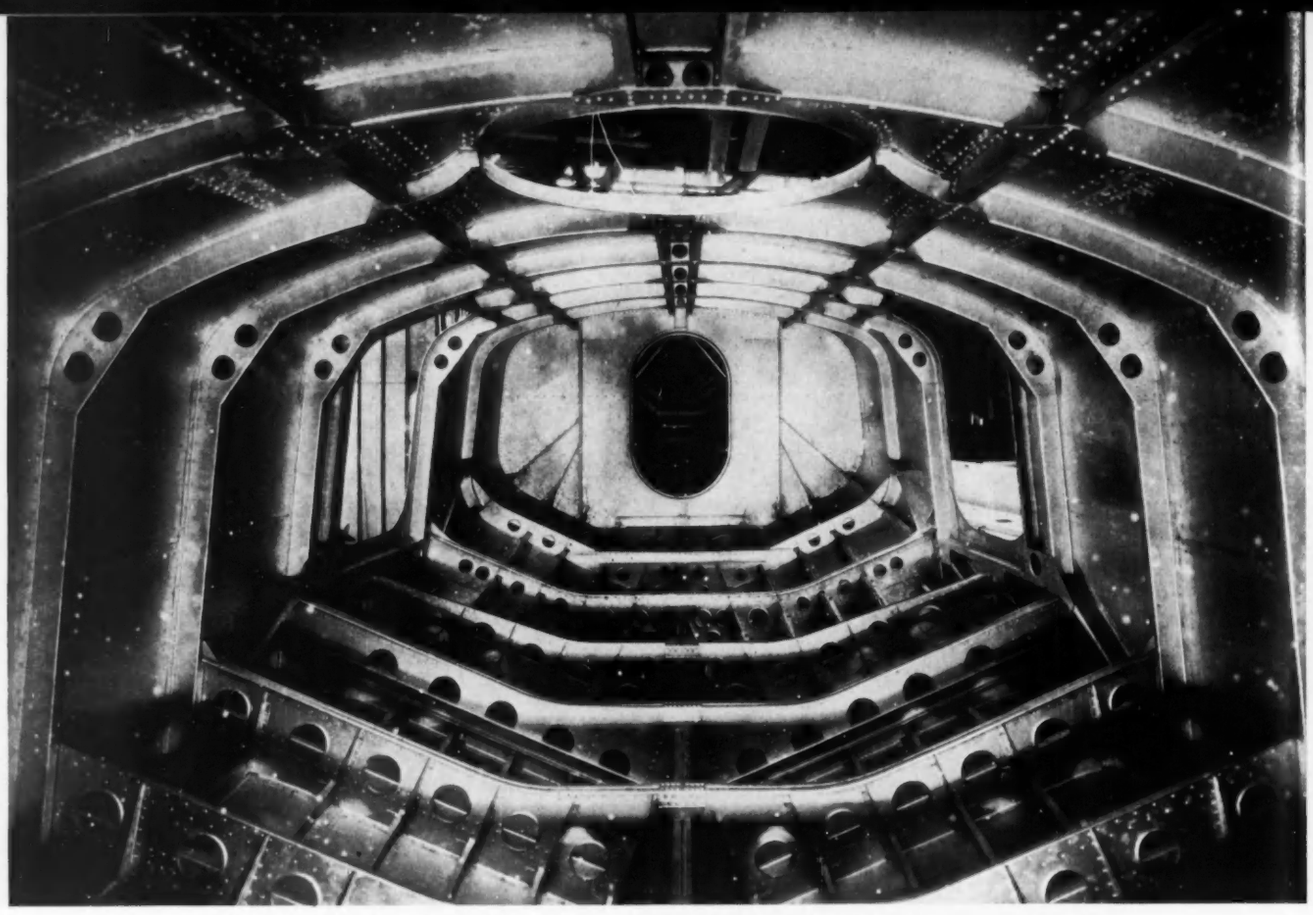


Fig. 5—Interior view of hull shown in Fig. 2. Note the Z-bar bottom stringers and the deep upper longitudinals which are used to reinforce the cutouts

rugations are covered with thin sheet, as is often done on airplane box beams, a coefficient of end restraint equal to 1.5 is used.

Assume that for the structure shown in Fig. 1 a smooth skin covering, reinforced with continuous fore and aft stringers were used instead of corrugations. In that case, if the applied bending moment does not buckle the sheet, we can still use Equation (1) for computing the bending stresses. However, let us assume that the curved smooth sheet buckles at a stress of 5,000 lb. per sq.in., but that the bending stresses which the structure must carry are considerably higher. For those conditions there are two methods of checking the strength of the structure. One is to use a fictitious allowable stress for the combination of curved sheet and stringers, computing the applied stress in accordance with the method outlined in the first part of this article. The other is to approximate the stresses which would actually be present in the structure for any design bending moment. The first method will give misleading results, as may be gathered from the following discussion, and since it is not based upon rational assumptions is apt to confuse the engineer who wishes to obtain a working knowledge of the subject.

If the moment of inertia of the section is based upon the fore and aft stringers and all of the curved sheet between stringers, a stress calculated by the equation $f_b = M\ y/I_x$ implies that the actual stress in the sheet between stringers is nearly the same as the stress in the stringers, since the only variable is y, the distance from the neutral axis. This is erroneous. Only up to that point at which the curved sheet does not buckle, is it correct to use all the sheet between stringers for the calculation of the moment of inertia. After the sheet has failed elastically it will hold its buckling load, and any applied bending moment greater than the one which produces buckling in the sheet will stress only the stringers. It is not claimed that the following method is absolutely infallible, but it will be seen that it is based on sounder reasoning than the conventional method and that the results obtained, so far as the applied stresses in the structure are concerned, will be in better agreement with the actual stresses determined in tests.

The proposed method is simply based upon an effective moment of inertia which takes into consideration the fact that, in an efficient semi-monocoque structure the sheet between the stiffeners is usually subjected to a much lower stress than the stiffeners. This will become apparent as we proceed with the example.

In order to compute the stress acting on a monocoque for which curved sheet and stringers are used, the first step is to determine the allowable stress of the sheet. For this purpose the method developed by E. E. Sechler (*A Preliminary Report on the Ultimate Compressive Strength of Curved Sheet Panels*) is used. The allowable load which a sheet reinforced by stringers can carry is equal to:

$$P = C\sqrt{ES_{ST}} \times t^2 \qquad (5)$$

where:

C = a coefficient dependent upon the parameters η and λ (Fig. 4)
E = modulus of elasticity of material
t = thickness of skin covering

It can be seen from Equation (5) that the allowable stress of the stiffener must be known before the sheet allowable can be determined. The stiffener used for this example is a hat section as shown in Fig. 6*a*.

Referring to the information in the article "Compression Members" (Feb. *P.E.*), the allowable stress is determined as follows:

Distance between frames = 20 in.
Coefficient of end restraint = 1.5 (assumed)
Radius of gyration, $\rho = 0.56$

$$L/\rho = \frac{20}{.56} = 35.7$$

$$B = \frac{45{,}000 \times 35.7^2}{1.5 \times \pi^2 \times 10{,}500{,}000} = 0.368$$

$$\sigma = \frac{45{,}000 \times 1.368}{1.368 + 0.368^2} = \frac{61{,}500}{1.504} = 41{,}000$$

$K_c = 13$ (assumed)

$$K = 13\left(\frac{45{,}000}{41{,}000}\right)^{\frac{1}{2}} = 13.6$$

$P/A = 41{,}000 \tanh 13.6 \times 0.040 = 20{,}200$ lb. per sq. in.

Having determined the allowable P/A of the stiffener, we can proceed with the determination of the sheet allowable.

$$\eta = \frac{b}{R}\sqrt{\frac{E}{S_{ST}}} = \frac{8.35}{42.5}\sqrt{\frac{10{,}500{,}000}{20{,}200}} = 4.48 \text{ (See Fig. 3)}$$

$$\lambda = \frac{t}{b}\sqrt{\frac{E}{S_{ST}}} = \frac{.032}{8.35}\sqrt{\frac{10{,}500{,}000}{20{,}200}} = 0.088$$

$C = 2.8$ (from Fig. 4)

$$\text{Sheet allowable} = 2.8\,(10{,}500{,}000 \times 20{,}200)^{\frac{1}{2}} \times \overline{.032}^2$$

$$= 2.8 \times 3245 \times 142.2 \times \overline{.032}^2 = 1{,}325 \text{ lb.}$$

$$\text{Allowable } P/A = \frac{1{,}325}{8.35 \times 0.032} = 4{,}860 \text{ lb. per sq. in.}$$

It should be noted here that unless the rivets attaching the stiffeners to the sheet are spaced sufficiently close together, the sheet value computed above will not be realized, since the sheet will buckle between rivets. Lacking test data pertaining to this subject, a rivet pitch equal to forty times the skin thickness is suggested as a safe limit.

The neutral axis and the moment of inertia of the section are calculated in the usual manner, except that a reduced area is used for that portion of the curved sheet which is subjected to compression stresses. The effective area is equal to:

$$A_{EFF} = A \times \frac{\text{sheet allowable}}{\text{stiffener allowable}} \times \frac{\text{dist. from extreme fiber to N.A.}}{\text{dist. from centroid of unit to N.A.}}$$

or simply,

$$A_{EFF} = A\,K$$

If the value of K should be greater than one, this latter value must be used.

It is apparent that because the sheet between stringers on the compression side is only partially effective, the true neutral axis of a symmetrical circular section no longer passes through the center of the circle. Consequently, since the effective areas, according to the values of Table IV in column 3, are in part based upon the ratio of $42.5/D$, where 42.5 is the distance between the center and the extreme fibers, and not the neutral axis and the extreme fibers, a slight error is introduced. Furthermore, the ratio of the sheet allowable to the stiffener allowable, used in Equation (6), should in fact be the ratio of the sheet allowable to the maximum fiber stress. These discrepancies are slight and the resulting errors negligible.

If the applied bending moment in the above example is equal to 3,300,000 in. lb., the maximum compression stress in the fibers would be:

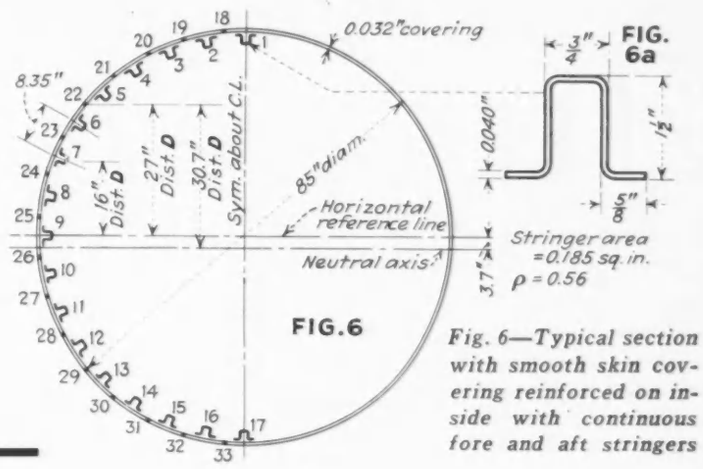


Fig. 6—Typical section with smooth skin covering reinforced on inside with continuous fore and aft stringers

$$f_c = \frac{-3{,}300{,}000 \times 45.9}{7{,}480} = -20{,}200 \text{ lb. per sq. in.}$$

As the allowable unit stress is equal to 20,200 lb. per sq.in. the unit stress as calculated is satisfactory.

Table IV—Computation of Neutral Axis and Moment of Inertia

(1) Unit	(2) Area	(3) D	(4) 42.5/D	(5) Eff. Area	(6) A D	(7) D_1	(8) $A D_1^2$
1	0.185	41.8	—	—	—	45.9	390
2	0.370	41.1	—	—	—	45.2	755
...	...	...	...	...	...	...	...
7	0.370	16.0	—	—	—	20.1	149
8	0.370	8.2	—	—	—	12.3	56
9	0.370	0	—	—	—	4.1	6
...	...	...	...	...	...	...	...
18	0.267	42.4	1.00	0.066	2.8	44.8	143
19	0.267	40.7	1.05	0.069	2.8	41.6	139
20	0.267	37.5	1.13	0.074	2.8	37.0	128
21	0.267	32.9	1.29	0.085	2.8	31.1	117
22	0.267	27.0	1.57	0.103	2.8	24.1	100
23	0.267	20.0	2.13	0.140	2.8	16.4	81
24	0.267	12.3	3.46	0.227	2.8	8.3	61
25	0.370	4.2	—	0.267	1.12	8.2	18
26	0.370	— 4.2	—	0.267	— 1.12	8.2	—
...	...	...	...	...	...	...	...
32	0.370	—40.7	—	0.267	—10.85	36.6	258
33	0.370	—42.4	—	0.267	—11.31	38.3	392
	10.192			10.192	—37.17		7,480

$$\text{Neutral axis} = \frac{-37.17}{10.192} = 3.7 \text{ in. below Reference line}$$

$I_{x-x} = 7{,}480$ in.

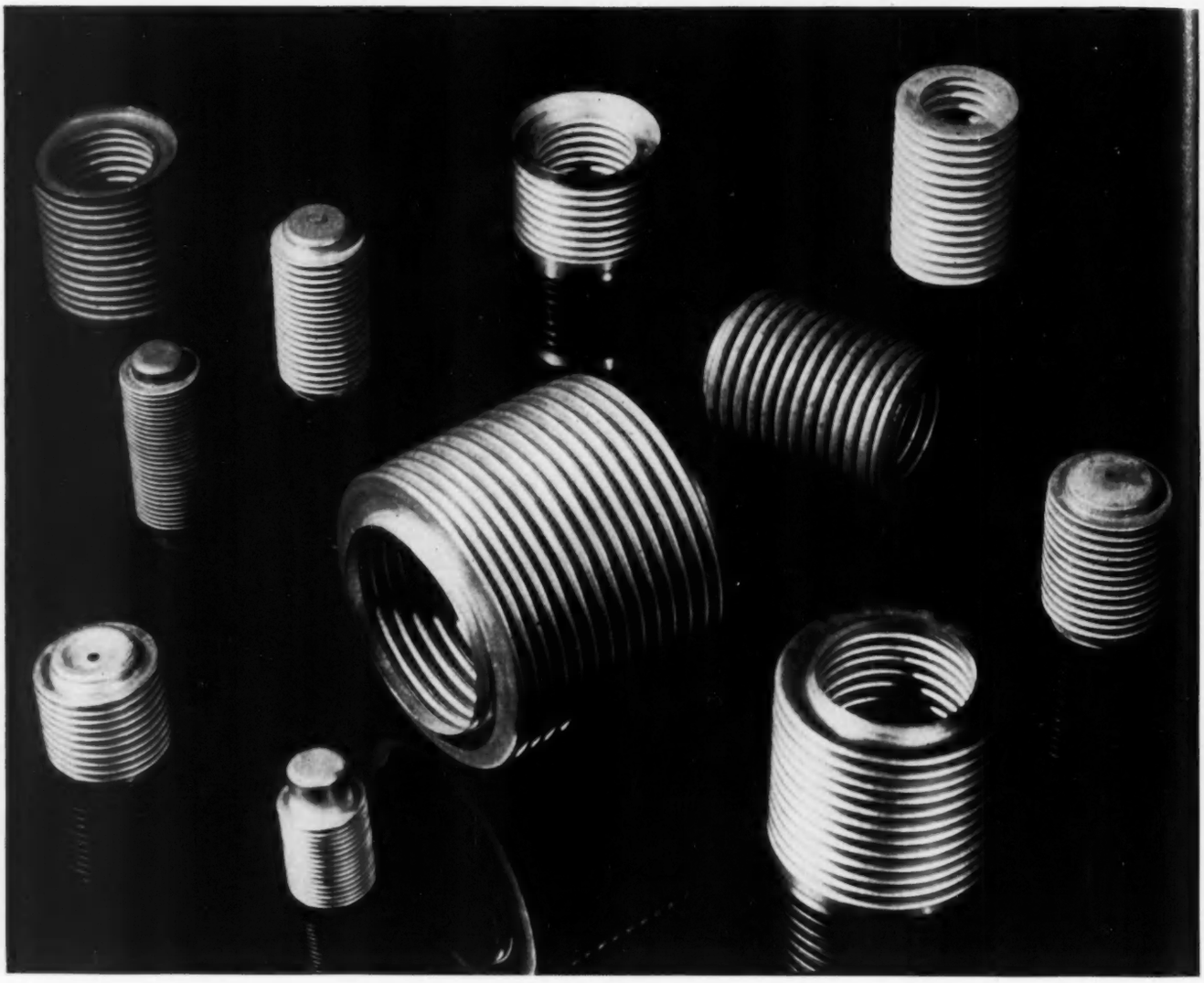
Text Reference

(1) A Preliminary Report on the Ultimate Compressive Strength of Curved Sheet Panels, by E. E. Sechler, California Institute of Technology—Publication No. 36.

Miscellaneous References

(1) "Skin Deep" by Joseph S. Newell, Aviation—Nov. and Dec. 1935.

(2) "Recent Aspects of Stressed Skin Construction" by Earle E. Blount, Journal of Aeronautical Sciences—Oct. 1934.



Steps in the manufacture of gas-tight flexible metal bellows of corrosion-resistant materials formed from drawn tubes by hydraulic pressure. Methods of designing for minimum cost require consideration of bellows manufacturing processes

When Selecting Bellows

WHILE serving as a weather forecaster in Knoxville, Tenn., in 1899, Weston M. Fulton was studying the fluctuations in pressure of air and vapor, and needed a method of confining gases that would allow them to expand and contract without escaping. Corrugated diaphragms failed at the soldered joints, and bellows made of leather and fabric could not be made gas-tight. It was then that Mr. Fulton turned to the development of a rolled bellows formed from a seamless drawn tube.

Essentially the same process as developed by Mr. Fulton is used today in the making of rolled bellows. By successive operations, a deep cup is drawn from a sheet to the desired length, diameter and wall thickness. Slight corrugations are then pressed into the wall, and in successive rolling operations these corrugations are deepened, one at a time, until the desired depth is obtained. For each rolling operation a different set of corrugating fixtures is needed for every combination of diameter and convolution depth. This method is relatively slow and expensive and except for large diameter bellows has been largely superseded by a hydraulic process.

In the hydraulic method, the same drawn tube used in the rolling process is expanded by internal pressure into grooves in the wall of the die, thereby forming the convolutions, the tube simultaneously shortening in length. Thus, the corrugating is performed rapidly in one operation and only one set of corrugating plates is required for each combination of diameter and convolution depth.

With the development of the low-cost hydraulic process, much wider applications of the metal bellows have been developed for pump diaphragms, flexible

seals, packless valves, expansion joints, and for thermostatic and pressure controls. Two of the largest users today are the automotive industry and refrigerator control manufacturers. But with greater attention being given to automatic control and operation in all types of machines, applications of bellows are being extended in many other fields of mechanical design.

In any application of a bellows unit, the space requirements, total movement, operating pressure, service temperature and corrosive influences must be known before the bellows can be specified. Because of the tool cost for an undeveloped design it is usually more economical to use a design of bellows having a convolution depth and outside diameter for which tools are already available.

The stroke obtainable from a bellows of given outside diameter increases directly with the number of convolutions and inversely with a power of the thickness. The deflection rate is proportional to the elastic modulus or stiffness of the metal. Expansion of the bellows per unit of pressure is constant for pressures throughout the working range, which is limited by the strength of the bellows material. Although design formulas are available for deflection-rate and stress, variations in the physical properties of the materials are so great that exact mathematical analysis is of little practical value. The final design must be checked by test.

Service life is also a factor in the selection of a given bellows. If space is limited, a higher strength material may be required to withstand a large number of operations. Since the metal is thin, even a small amount of corrosion will greatly reduce the life. No surface finishes give complete protection hence a material must be selected that will itself resist attack.

To prevent the possibility of overstress, stops must be incorporated in the design of the bellows or the operating mechanism to limit travel and thereby limit the amount of stress. The simplest method is to have the bellows operate by compression rather than by extension, overtravel being prevented by the bellows closing against an internal stop. Without the stop the bellows will close solid, tending to flatten the convolutions and develop high stresses.

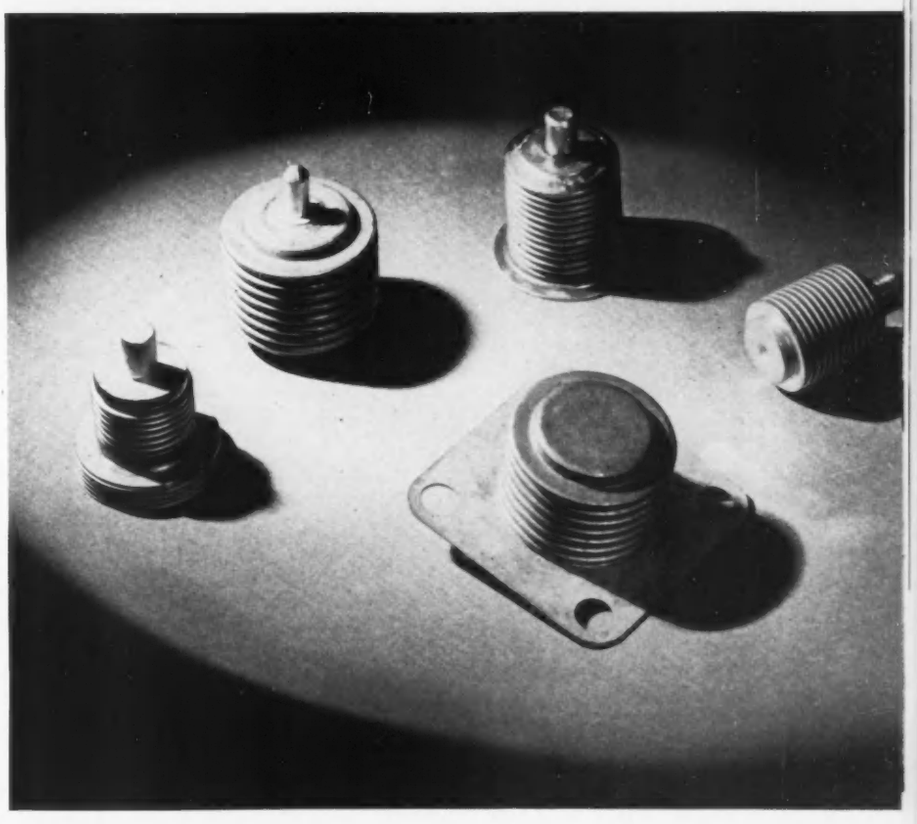
Both operating pressure and temperature have an important bearing on the design of bellows. Excessive pressure will not only overstress the material but will also cause distortion of the bellows. By using a heavier wall thickness, higher pressures can be applied safely but the increased thickness greatly increases the stiffness. Likewise, a smaller ratio of convolution depth to diameter permits higher pressures but also reduces the flexibility. Where both flexibility and resistance to high pressure are needed, bellows can be made of two-ply construction by fitting one tube inside another and forming both in the same operation.

Operating temperatures are limited by two factors—the properties of the material and the strength of the solder joints usualiy needed to attach fittings. Working stresses and allowable deflections must be reduced when temperatures are higher. In attaching heads and fittings soft solders are used ordinarily, since the soldering temperature is below the annealing point of the material. Where silver solder or welding is necessary the beneficial effects of cold working resulting from the forming operations can still be retained by special heating technique, but this involves added cost.

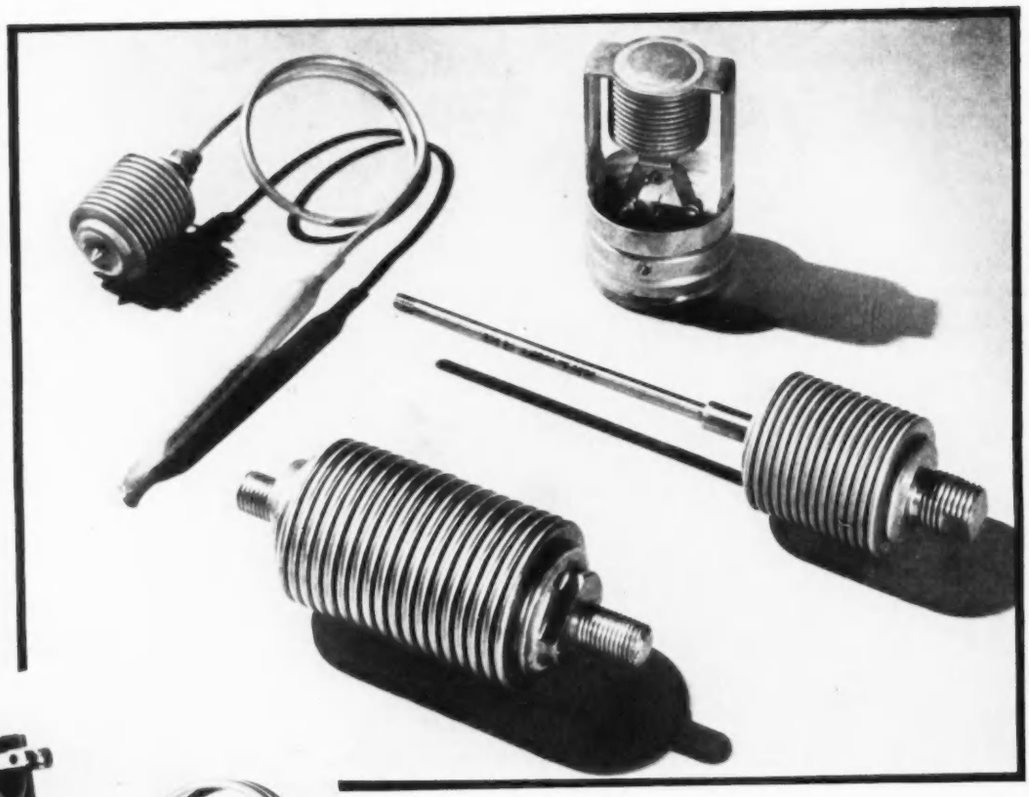
In selecting the bellows material, ease of forming, material cost and corrosion resistance are the most important factors. Because of its resistance to atmospheric corrosion brass is used in most applications. For contact with oil, alcohol and most refrigerants brass readily resists attack. Also the material is easy to draw and can be formed to close tolerances. However, brass cannot be used in contact with mercury, ammonia, brine, certain types of oil, water vapor and water containing some kinds of boiler compounds.

For applications wherein brass lacks sufficient strength, phosphor bronze can be used. It has the added advantage of withstanding higher temperatures than can be used with brass. Bronze is somewhat more

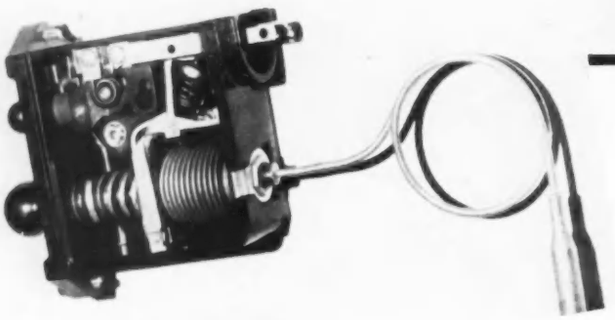
The assistance of the engineers of the following companies for the contributions of design data and information is acknowledged:
Bridgeport Brass Company **Fulton Sylphon Company**
Clifford Manufacturing Company



Ends of bellows are closed pressure tight by soldered fittings or by closed end of drawn tube



Thermostatic elements for a refrigerator control, an automobile radiator thermostat, an airplane engine oil temperature regulator, and a tractor cooling water control



In refrigerators, bellows units are used both for the temperature - sensitive control element and for a gas-tight compressor seal

resistant to atmospheric corrosion but is more difficult to work.

Monel metal is being used more widely for bellows, particularly to resist ammonia. This material not only has high corrosion resistance but also has high strength and retains its properties better at elevated temperatures than brass or bronze. However, Monel metal is difficult to draw and form, and this factor in addition to the higher price of the metal increases the cost of the complete bellows.

Other materials can also be used in bellows to meet special service conditions. Heat-treated beryllium copper allows the use of higher working stresses. Silver is used where the bellows is in contact with certain chemicals such as chlorine.

Steel is also being used for bellows, particularly for resisting corrosion from ammonia and mercury. Although it is difficult to draw thin uniform-wall steel tubing to close tolerances in diameter and wall thickness, commercial processes are being perfected and suitable steel tubing soon will be available. Another difficulty in using steel has been that corrosion will quickly damage the thin wall required for flexibility. Work has been done on developing processes for making stainless steel thin-wall tubing and such tubes are now available.

To keep down costs, bellows of standard diameter, convolution depth, and wall thickness should be used whenever possible. A special bellows required by some limitation imposed by the design may involve added cost and delay in making tools for forming, and further delay in obtaining tubing of special diameter or wall thickness.

As an illustration of the method of using a modification of a standard design, suppose that a bellows is needed for a pressure relief valve in a hot water system. The operating pressure is 8 lb. per sq.in., and an operating force of 8 to 10 lb. is required with a movement of $\frac{3}{16}$ in. From data supplied by the bellows manufacturer, a bellows with an outside diameter of $1\frac{1}{2}$ in. will give the required operating force. The low service pressure permits the use of a thin wall of about 0.005 in. which still allows a good factor of safety for corrosion. Since the bellows will be in contact with water vapor, bronze will be more suitable than brass for such an installation.

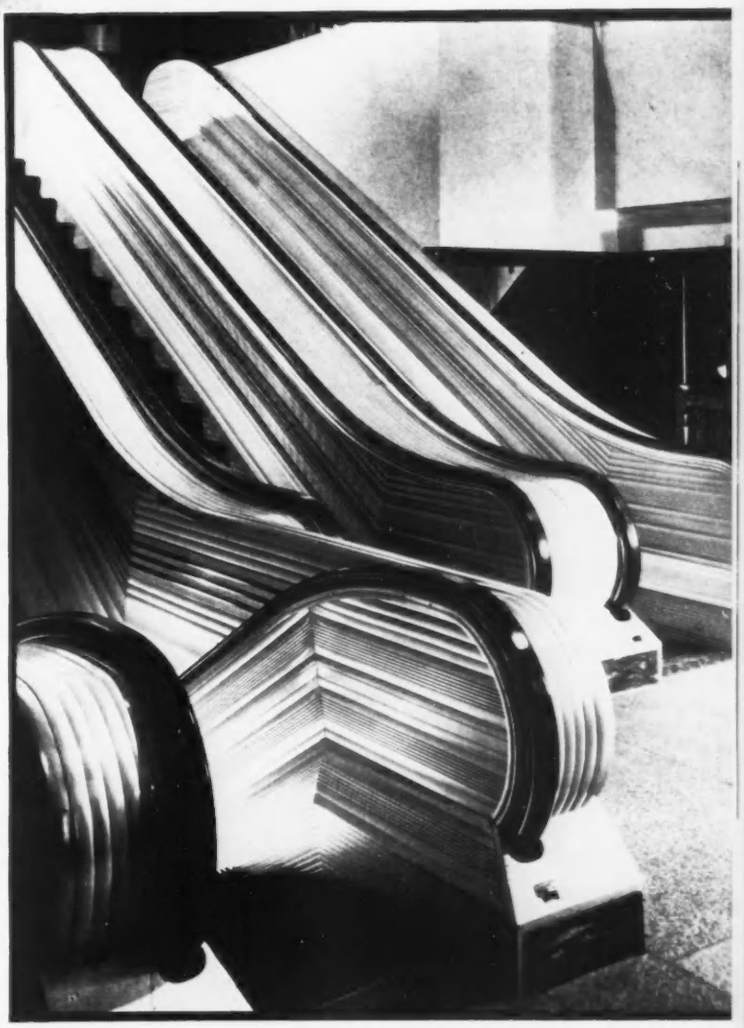
For the $\frac{3}{16}$ in. movement, manufacturer's data show that the $1\frac{1}{2}$ in. diameter bronze bellows will require 13 convolutions if space is available. If the length is restricted, 10 convolutions will do, but the working stress will be higher. Adding to the length of the 13 convolutions the length of shoulders and spun ends gives a total length of $2\frac{1}{8}$ in. for the bellows. If a shorter length is necessary, a larger diameter will give a larger stroke per convolution and therefore fewer convolutions will be required, hence a shorter length

ALLOYS of ALUMINUM

PAUL V. FARAGHER
Aluminum Company of America

PART II

Types of aluminum casting alloys and the workability, riveting and heat-treatment. Part I appeared in February



(Above) Aluminum parts for Toledo Plaskon scale. (Right) Application of extruded aluminum alloy shapes finished by the Alumilite process

FOR the production of aluminum alloy castings, an alloy containing approximately 8 per cent copper was the first to be used in quantity. It was called "No. 12" and is still widely used, although the modifications 112 and 212, in which the limits for impurities are different, and certain elements are added in small amounts to improve the casting or machining qualities, are more frequently specified at the present time. These alloys have fair mechanical properties and are suitable for many purposes for which the service requirements are not too exacting.

For castings of relatively thin sections which are to be exposed to corrosive conditions, such as are encountered in architectural and marine applications, alloy 43 containing 5 per cent silicon is better suited. Sound leak-proof castings in this alloy are easily produced and are now widely used. The strength of 43 alloy is slightly lower than that of alloy 12 and the related alloys, but its elongation is appreciably higher. It is also not quite so easy to machine, but no serious difficulties are experienced if proper tools and machining practices are used.

Alloy 356 containing slightly more silicon and a small amount of magnesium may be heat-treated to develop

distinctly higher mechanical properties. Castings of this alloy offer the advantages mentioned for alloy 43, and in addition, have high strength. The heat-treatment may be varied to meet the requirements of the service which the casting must perform.

The oldest of the heat-treated casting alloys and the one which is still generally used is alloy 195. In the heat-treatment designated by the symbol "T4," this alloy has its maximum ductility and resistance to impact stresses, together with good tensile and yield strength. On standing at room temperature it age-hardens and the properties approach those of 195-T6, which differs from 195-T4 in having a somewhat higher tensile strength, distinctly higher yield strength, and an elongation about half as great. Another variation in heat-treatment produces 195-T62, which is still harder and stronger than 195-T6, but because of its low elongation it is not specified for purposes requiring the maximum resistance to severe shocks.

Alloy 214 is specified for castings which must have the highest possible resistance to corrosion under severe conditions of exposure. It is more difficult to cast into intricate castings than some of the other alloys and for certain designs it may be necessary to substitute B214 alloy, a modified composition having superior casting qualities and resistance to corrosion almost equal to that of the parent alloy, although somewhat lower physical properties.

Table V—Approximate Radii for 90° Cold Bend of Aluminum and Aluminum Alloy Sheet

Minimum permissible radius varies with nature of forming operation, type of forming equipment, and design and condition of tools. Minimum working radius for given material or hardest alloy and temper for a given radius can be ascertained only by actual trial under contemplated conditions of fabrication.

Alloy and Temper	Bend Classification*	Alloy and Temper	Bend Classification*
2S-O	A	17S-RT(1)	J
2S-¼H	B	A17S-O	B
2S-½H	B	A17S-T	F
2S-¾H	D	24S-O(1)	B
2S-H	F	24S-T(1)(2)	J
3S-O	A	24S-RT(1)	K
3S-¼H	B	51S-O	A
3S-½H	C	51S-W	F
3S-¾H	E	51S-T	K
3S-H	G	52S-O	A
4S-O	B	52S-¼H	C
4S-¼H	D	52S-½H	D
4S-½H	F	52S-¾H	F
4S-¾H	H	52S-H	G
4S-H	J	53S-O	A
17S-O(1)	B	53S-W	F
17S-T(1)(2)	H	53S-T	G

*For corresponding bend radii see following table:

Radii Required for 90° Bend in Terms of Thickness, t

		Approximate Thickness					
B & S Gauge		26	20	14	8	5	2
Inch		0.016	0.032	0.064	0.128	0.189	0.258
Inch		1/64	1/32	1/16	1/8	3/16	1/4
Bend Classification	A	0	0	0	0	0	0
	B	0	0	0	0	0-1t	0-1t
	C	0	0	0	0-1t	0-1t	½t-1½t
	D	0	0	0-1t	½t-1½t	1t-2t	1½t-3t
	E	0-1t	0-1t	½t-1½t	1t-2t	1½t-3t	2t-4t
	F	0-1t	½t-1½t	1t-2t	1½t-3t	2t-4t	2t-4t
	G	½t-1½t	1t-2t	1½t-3t	2t-4t	3t-5t	4t-6t
	H	1t-2t	1½t-3t	2t-4t	3t-5t	4t-6t	4t-6t
	J	1½t-3t	2t-4t	3t-5t	4t-6t	4t-6t	5t-7t
	K	2t-4t	3t-5t	3t-5t	4t-6t	5t-7t	6t-10t

(1) Alclad 17S and Alclad 24S can be bent over slightly smaller radii than the corresponding tempers of the uncoated alloy.

(2) Immediately after quenching, alloys can be formed over appreciably smaller radii.

High-Strength Castings

Castings of alloy 220-T4 have the highest mechanical properties and resistance to abuse of any of the aluminum casting alloys. This alloy has been used for the production of railway car trucks, for parts of large shovel dippers, for certain aircraft parts and other applications requiring maximum strength.

Many more alloys of aluminum are used for castings for special purposes. Some like 122, 132, 143, 355 and A355 are used where high temperatures are encountered in service because they retain their mechanical properties well under these conditions.

The mechanical properties shown in Table IV (*Feb. P.E. p. 64*) are those obtained from standard tension test specimens separately cast in sand. The test specimen obviously cannot be a direct measure of the strength of the casting. When properly poured under controlled conditions it does serve as a measure of alloy quality and as a means of controlling heat-treatment processes and as an indication of the ability of the producer to apply proper foundry control. When used with the factors of safety ordinarily applied in engineering practice, the mechanical properties specified for the alloys may be used with every expectation of satisfactory performance in a casting which is properly designed.

Forming Operations

Aluminum alloys are fabricated in the same types of tools as are used in forming other metals. One of the outstanding qualities of pure aluminum is its high degree of workability. As the strength of the metal is increased by the formation of alloys, there is some loss in this quality, although the forming qualities cannot be compared directly on the basis of relative tensile strengths. The data in Table V can be considered as only a guide in arriving at a proper choice of alloy and temper or of the radius for forming tools.

Outdoor meter box die-cast from aluminum alloy

The final answer can be obtained only by trial of samples under the conditions which are to be used in production.

In forming alloys like 17S-T or 24S-T, somewhat larger radii are required for bends than would be used for a similar thickness of steel. For some operations it may be necessary to heat-treat the alloy and form it within an hour or two after it has been quenched, before it has age-hardened. The interval between quenching and forming may be prolonged if facilities are available for keeping the metal at the temperature of melting ice or lower, which decreases the rate of age-hardening. For some operations it is necessary to form these alloys in the soft temper and subsequently heat-treat the formed article. In the case of some of these aluminum alloys it is readily possible to improve their forming qualities by heating the material and working it at a somewhat elevated temperature.

Alloys 27S-T and 53S-T may be heated at a temperature as high as 400 deg. F. for about a half hour without appreciable effect upon their physical properties or corrosion resistance. At this temperature the increase in plasticity is sufficient to make possible many forming operations which could not be carried out at room temperatures. Care must be exercised not to raise the temperature too high or hold it for too long a time; otherwise the properties may be impaired when the metal is subsequently cooled. This process may also be safely used with 2S, 3S, 4S and 52S alloys.

Heating 17S-T and 24S-T under these conditions has little effect on their mechanical properties, but it does seriously decrease their resistance to corrosion. Consequently the method is not applicable to them. However, it is possible to heat them to the heat-treatment temperature and form them in tools having a sufficient volume of metal to cool them rapidly during the process of forming. Under these conditions the rapid cooling in the tools constitutes the quench and the alloys develop the properties of the heat-treated temper on age-hardening at room temperatures (U. S. Patent 1,751,500). This method is obviously most readily adapted to the forming of heavier sections. It has the advantage that there is no warping such as might result if the section were formed from 17S-O or 24S-O and subsequently heat-treated and quenched in water.

Four steps in the production of the part shown in the lower right, involving impact extrusion and drawing

Riveting

Rivets of the commonly used types are available in several different alloys, the choice among them depending upon the nature of the work and the materials to be joined. For riveting 2S or 3S, rivets of the same alloy are commonly used; for 52S, 53S and Alclad sheet, 53S rivets are usually specified. In sizes up to about $\frac{3}{8}$ in. these rivets may be readily driven in the full heat-treated temper, 53S-T; larger sizes should be used in the "as quenched" condition, 53S-W. The design should, of course, be based on the shearing strength of the alloy in the temper which is used.

The choice of rivets to be used with 17S-T, 24S-T and 27S-T depends upon a number of considerations. The use of 17S rivets requires facilities for their heat-treatment. Sizes up to about $\frac{3}{8}$ in. may be driven cold, provided they are used within about one hour after they have been quenched. If they are stored at the temperature of melting ice or lower, the age-hardening is retarded so that larger quantities may be heat-treated at one time.

Larger sizes should be driven hot. The rivets are heated to the heat-treatment temperature and are headed with minimum delay after removal from the furnace. The heat is removed from the rivet by the heading tools and by the metal in which it is driven, with sufficient rapidity to constitute quench, so that the properties of 17S-T are developed as the rivet age-hardens on standing.

For large structures requiring larger sizes of rivets, it may be necessary to use hot steel rivets, following the practice used for steel structures. Where smaller sizes are suitable, A17S-T rivets may be used. This alloy, a modification of 17S, having lower mechanical properties, is more easily worked and rivets in sizes up to about $\frac{1}{2}$ in. can be driven cold as they are received from the manufacturer. The lower shear strength of A17S-T as compared with 17S-T may be compensated in the choice of the size and spacing of the rivets.

Squeeze riveters are preferred for driving aluminum alloy rivets because they work the metal more uniformly. Pneumatic hammers are suitable but they should be of sufficient capacity to form the heat with a relatively small number of blows, so as not to harden the metal near the surface before the working has penetrated to the interior of the head. For cold heading 17S rivets, the required capacity of the riveter must be greater than that for hot steel rivets of the same size.

Heat-Treatment and Annealing

Although usually aluminum alloys are obtained from the manufacturer in the tempers in which they are to be used, there are a few occasions in which it may be desired to heat-treat or to anneal the metal to facilitate forming which must be done. While these operations

are readily accomplished, some care is necessary to obtain the best results. Detailed instructions may be obtained from the manufacturer for carrying out the operations which may be required.

For 17S alloy, the heat-treatment temperature is 940 deg. F., and for 24S it is 920 deg. F., with a permissible variation of plus or minus ten degrees. If these limits are not observed, inferior results may be obtained either from incipient melting of the alloy or from incomplete heat-treatment. The metal is heated at the specified temperature for a sufficient time to insure solution of the hardening constituents and is then promptly quenched in cold water.

Aluminum alloys may be annealed by heating them to a temperature around 650 deg. F. To obtain complete softening of 3S alloy, a temperature about 100 deg. higher is required. Since the use of too high temperature when annealing the heat-treatable alloys will produce some heat-treatment effects, more careful temperature control is necessary. The effects of strain hardening may be removed by heating in the range of 650 deg. F. to 670 deg. F.; this treatment also removes most of the effects of heat-treatment and is sufficient for many forming processes. To remove the effects of heat-treatment completely, it is necessary to heat the alloys at a temperature of 850 deg. F. for a short time and then cool them very slowly.

▼ ▼ ▼

ROBOTYPER

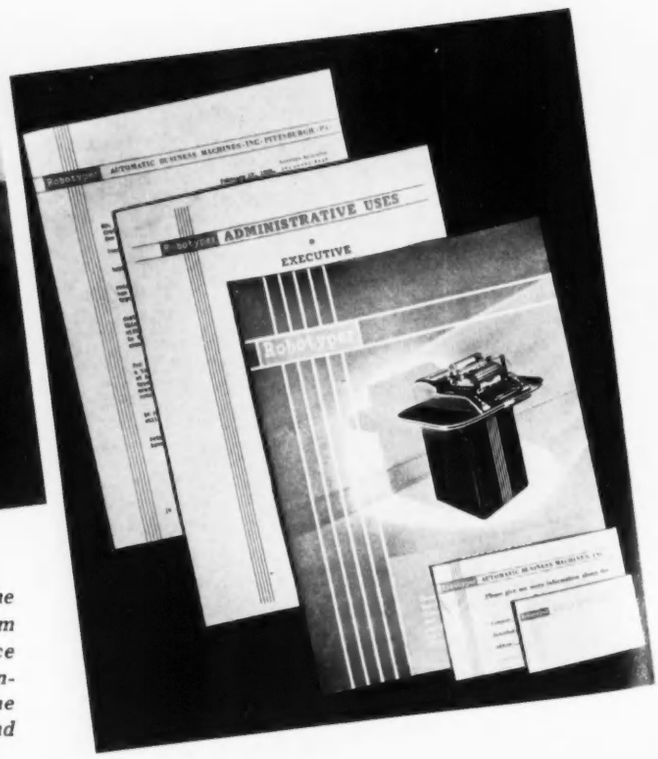
ORIGINALLY typed letters are written on a standard typewriter at three times the speed of an expert typist, with every letter personalized with name, address, salutation and desired changes in the body of the letter. Yet the operator still has time for other stenographic or clerical duties.

How is it done? In the Robotyper machine shown here, built by Automatic Business Machines, Inc., the typewriter keys are operated by a vacuum bellows controlled by perforated paper record rolls similar to player piano rolls.

The paper record rolls are perforated on a simple machine with a standard typewriter keyboard. Then the roll is placed in the mechanism built into the Robotyper desk at the rear of the typewriter. This mechanism or control unit governs all operating functions of the typewriter such as carriage return, rewinding and repeat, in addition to typing the body of the letter. An automatic switch stops the machine if the operator is busy with other duties when a letter is completed or fill-ins are to be made manually.

To reduce weight an aluminum framework is used for the mechanism. Bearings are of graphite-bronze, are self-adjusting and require no lubrication. Leather is used in the action diaphragms, and power is supplied by a small motor-driven vacuum pump built into the unit. Only one diaphragm is employed to operate all type bars, thereby giving identical pressure on all keys.

Although the mechanism itself is small, it was desired to make the product a self-contained complete unit. Therefore, a desk was designed with necessary storage space for stationary and paper rolls and to provide a working surface. To give the complete unit distinction, Otto Kuhler was commissioned to develop the appearance phase of the design. As shown in the accompanying picture, the vertical bands are aluminum, with the name plate on a red background at the intersection of the horizontal and vertical lines.



Designed by Otto Kuhler, this desk for the Robotyper houses the operating mechanism for the typewriter and provides storage space with minimum floor space. (Right) Horizontal and vertical lines centering on the name plate are carried over into letterheads and printed matter

How to Apply V-Leather Packings

J. N. SMITH
E. F. Houghton & Company

Advantages of V-leather, the factors that determine the thickness, proportions and numbers to use and how to support them in the stuffing box

JUST as a cup, flange or U-type packing has its own special features from the standpoint of economical operation, practical application and ultimate life, so does a leather V-packing have certain distinct advantages not obtainable in any other type of mechanical packing. These advantages arise mainly from the wedge-like shape of the V-packing, the method of supporting it and the manner in which the packing operates as a seal.

When properly supported, only the lip of the packing is in contact with the plunger or rod, as contrasted with full side-wall contact as in other types of packings. And whereas in these other types of packings the shoulder will wear first and failure occurs at that point, with V-packings the metal supporting rings, as shown in Fig. 1, protect the shoulder of the packing.

In addition to its inherently long wear life for the reason as given above, the V-packing by virtue of its wedge-shape and the design of the supporting rings can be put under lateral compression. By tightening the packing gland ring the V of the packing is spread as indicated in Fig. 1. This action causes the lip of the packing to exert a postive pressure against the wall of the packing housing and the surface of the ram or plunger, serving as a compression packing.

It is this feature of operating as a compression packing that gives the V-packing many of its advantages. Thus, after a set of leather V-packings has been in service to the full extent of its normal life as a mechanical packing it can be made to continue to function as a compression packing by taking up on the gland ring. This is a most important consideration in machines where shut-downs for packing replacement would incur a large production loss. In such applications the V-packings can be made to stop the leakage by drawing up on the gland ring, holding the pressure for a number of hours by the compression of the packing. Thus, repacking of the press can be delayed until such time as the production schedule will permit doing so without incurring lost time.

Another application wherein V-packings excel by virtue of their inherent characteristics is in packings for hydraulic presses with cast iron plungers or rams. It is extremely difficult to pack such a press with any type of mechanical packing other than the V-leather and obtain a reasonable degree of service. The highly abrasive character of the cast iron will wear a leather packing rapidly. In spite of the degree to which it has been polished, with water as the medium, the plunger will pit. The resulting rough surface will wear and tear any type of packing. But with properly supported V-leather packing only the lip of the packing is in contact with the plunger and hence the amount of wear is less. In addition, the ability of the V-leather to serve as a compression packing adds to its life.

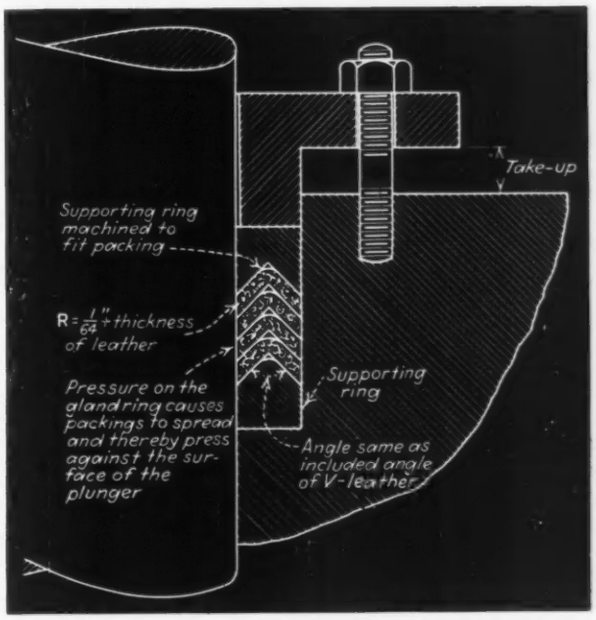

Take-up
Supporting ring machined to fit packing
R = 1/64" + thickness of leather
Pressure on the gland ring causes packings to spread and thereby press against the surface of the plunger
Supporting ring
Angle same as included angle of V-leather

Fig. 1—Typical stuffing box and packing design showing how the V-leathers maintain a flexible and long wearing seal

Table I—Thickness of Leather for V-Packings

Piston Diameter	Leather Thickness	
	45 Deg.	30 Deg.
3/8 in.	3/32 in.	1/16 in.
1/2 in.	1/8 in.	3/32 in.
5/8 in.	5/32 in.	1/8 in.
3/4 in.	3/16 in.	5/32 or 3/16 in.
7/8 in.	3/16 in.	3/16 in.
1 in.	3/16 in.	3/16 in.

Similarly, leather V-packings have also proved their outstanding merit on rotating shafts or plungers. Here also the lip contact feature enables them to withstand the continuous wearing caused by the friction of rotation. They operate as a mechanical packing, but there is also the distinct advantage of being able to take up on the gland ring after excessive wear has occurred, thereby spreading the V and effecting a seal.

Another installation wherein exceptional service can be expected of V-leathers is on plungers that are loaded

eccentrically or are subjected to the eccentric motions of the supported unit. Such continual loading or motion may cause the plunger to become out-of-round. As a result, the conventional leather packing cannot press against the plunger with a uniform pressure at every point in the circumference. But V-packings have such an exceptional degree of flexibility that they will compensate for such irregularity.

Similarly, presses will sometimes be found to have excessive clearance between the plunger and the cylinder wall or between the plunger and the gland ring, causing a premature failure of the packing. This condition can be eliminated in most instances by the use of V-packings. Properly supported by the necessary metal supporting rings having the proper clearance, the design of the V-packing can be made to compensate for the excessive plunger clearance and there will be no danger of packing being pinched in the clearance space.

In designing to use V-leather packings two factors must be determined. These are the cross-sectional dimensions of the packing to be used, and secondly, the number of rings to be used. With reference to the latter, there must be at least enough rings so that the total clearance between the rings will permit enough axial take-up to get the desired spread. At least three rings should be used, the average is 4 to 6 for pressures up to 15,000 lb. per sq.in. In general, the greater the pressure and speed, the greater the number of rings. In special cases such as when the packing also serves as a bearing as in some horizontal presses, additional rings must be provided. The minimum number of rings, three, should be used when pressures are low, in order to minimize frictional losses. But high plunger velocities, even at low pressures, may require more rings.

The selection of the packing dimensions not only for service requirements, but also as to the limitations of molding leather, has been developed through experience. The pressure and the type of motions will determine not only the number of packings but also the thickness of the leather to be used. The angle of V may be 30 deg. or 45 deg., although many manufacturers use a lesser or greater angle. For a given leather, a greater angle will necessitate a wider packing space. In the accompanying Table I are given the recommended thickness of leather for V-packings for different piston or plunger diameters, and in Table II are given the principal dimensions of V-packings of various thicknesses of leather and for both 30 deg. and 45 deg. angles. The recommended dimensions are those within the heavy lines.

Both top and bottom supporting rings of metal are recommended as they present a more rigid and economical installation. Metal is not only better able to withstand the pressure to which it may be subjected, but it also can be used over and over again. Leather or composition supporting rings can be used, but usually they must be renewed with each replacement of the packing. Furthermore, if composition supporting rings are to be used the type of hydraulic medium must be taken into account. For example, oil has a great tendency to disintegrate rubber compositions.

Supporting rings should be machined to the same included angle as the angle of the packing, as shown in Fig. 1. This will give the proper support and flexibility. The supporting ring that is to fit into the V-section of the packing, shown as the upper ring in Fig. 1, should be machined with a radius equal to the radius at the base of the packing.

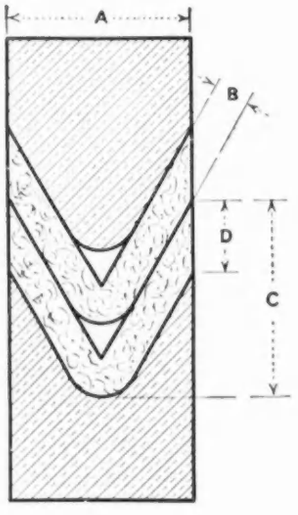
The two important requirements for the success of any type of packing design are to properly support the packing and to maintain proper clearances. With a set of V-packings another factor should be given consideration and that is the angle of the supporting rings. Experience has proved that supporting rings made on an angle different from that of the packings, do not give the best results.

Leather V-packings are not a "cure all" for packing design. However, they do have a specific place in industry, and for certain applications and service conditions excel all other types.

Table II—Recommended Proportions for Leather V-Packings

Recommended dimensions are within heavy lines. Dimensions in inches.

A Width	B Thickness	45 deg. Angle		30 deg. Angle		R Radius Ring
		C-Depth of Packing	D-Depth in Contact	C-Depth of Packing	D-Depth in Contact	
1/4	1/16	0.172	0.088	0.263	0.125	5/64
3/8	1/16	0.234	0.088	0.372	0.125	5/64
3/8	3/32	0.266	0.133	0.403	0.187	7/64
1/2	3/32	0.328	0.133	0.511	0.187	7/64
1/2	1/8	0.359	0.177	0.542	0.25	9/64
5/8	1/8	0.422	0.177	0.650	0.25	9/64
5/8	5/32	0.453	0.221	0.682	0.312	11/64
3/4	5/32	0.516	0.221	0.790	0.312	11/64
3/4	3/16	0.547	0.265	0.821	0.375	13/64
7/8	5/32	0.578	0.221	0.898	0.312	11/64
7/8	3/16	0.609	0.265	0.930	0.375	13/64
1	3/16	0.672	0.265	1.038	0.375	13/64
1 1/8	3/16	0.734	0.265	1.146	0.375	13/64
1 1/4	3/16	0.797	0.265	1.254	0.375	13/64
1 3/8	3/16	0.859	0.265	1.363	0.375	13/64
1 1/2	3/16	0.922	0.265	1.471	0.375	13/64



Specifications for Painting and Lacquering

Standards Division—Materials and Process Engineering Department
Westinghouse Electric & Manufacturing Company

IN THE first section of these specifications on page 464 in the December number, information of a general nature was given for surface preparation and application of the paint. In this section are given detailed specifications for painting conditions and two types of finishes for steel parts.

PROPER PAINTING CONDITIONS: Proper ventilation, uniform temperatures, well kept equipment, and good light are all essential to obtaining good results. Paint dries slowly in cold or damp locations. Light is most important and should be strong but free from glare. Strong air currents blowing over the painted surfaces are to be avoided.

It is absolutely necessary to have oil and water separators on all air lines connected with spray equipment.

All portable tank spray equipment containing pigmented materials must be equipped with power driven agitators which are permitted to operate continuously. When hand spray cans or open paint cans are used, the operators shall be instructed to stir the contents thoroughly with a flat paddle at least once an hour. Pigmented paints settle out rapidly after thinning.

SPECIAL INSTRUCTIONS FOR LACQUERING: Lacquer should be applied by spraying. Do not use brushes, as brush application is very difficult and is practical only on small parts or on touch up work. When brush application is necessary, it should be flowed on, using a soft hair brush.

Lacquers are more susceptible to poor painting conditions and improper cleaning methods than any other type of material. The grease deposited on smooth metal parts by touching with the hands is sufficient to injure the adhesion, and may cause the lacquer to peel off after it is dry. The surface to be lacquered must be as nearly chemically clean as possible and then lacquered at once. When trouble with peeling is experienced, the parts should be washed in clean lacquer thinner and wiped dry with clean cloths.

Smooth polished metal surfaces tax the adhesion of any finishing material. If possible, such surfaces should be roughened by shot blasting or rubbing with abrasive cloth.

Blushing or whitening of the lacquer film is caused by moisture condensing on the surface of the lacquer film while it is drying and causing the solids in the lacquer to throw out of solution. The addition of about 10 per cent of butyl acetate to the lacquer thinner will usually prevent blushing on wet days unless drafts are permitted to blow on the lacquered surface, or if moisture is present in the air line.

Pitting and blistering of the finish is usually caused by the lacquer drying too fast, thereby trapping air bubbles in the film. It can likewise be overcome by using retarder and by eliminating drafts over the work.

The following points should be observed to prevent most of the troubles in lacquering:

1. Prevent drafts from open doors and windows from striking the work. Close windows near the spray booth and cut exhaust fan speed to avoid drafts over the work.

2. Keep the lacquer room shut off from rooms which contain acid tanks, cleaning baths, or plating tanks giving off steam and moisture.

3. Preheating the work before lacquering is helpful. The temperature of the object should not exceed 40 deg. C. when lacquer is applied or a pebble-like finish will be obtained.

4. Keep moisture separators in the air line in good operating condition and drain frequently.

Two-Coat Inhibitive Prime and Lacquer Finish

This is a two coat finish consisting of one coat of a durable gloss black lacquer applied over an inhibitive primer. It is especially suited for steel or iron parts, but may be applied over cadmium plate, galvanizing, and others.

CLEANING: Clean surfaces thoroughly. Remove rust and loose scale by shot blasting, or by scraping and wire brushing. Remove oil, dirt, and grease thoroughly, using a solvent wash or alkaline cleaning methods. Powder deposits left on the surface after cleaning shall be removed thoroughly by wire brushing and wiping or blowing.

PRIMING: As soon as possible after cleaning, apply a uniform coat of red priming paint thinned with paint thinner to the correct consistency. Air dry until hard, approximately 4 hr., before applying the next coat, or bake at 230 deg. F.

LACQUERING: Apply a uniform coat of gloss black lacquer thinned with lacquer thinner to the correct consistency, not more than one part thinner to one part lacquer being required. Air dry until hard before handling, from 15 to 30 min. being required.

Three-Coat Air Drying Paint for Outdoor Exposure

This is a durable, three coat, dark gray, glossy, air-drying finish. It is applied mainly to steel tanks and structural steel parts subjected to outdoor exposure.

CLEANING: Remove rust and scale thoroughly by sand or shot blasting if possible. Otherwise use scraping and wire brushing. Grease, oil and dirt must be removed thoroughly.

PRIMING: Apply all paints by brushing, spraying, flowing or dipping. Prime all surfaces with primer thinned with paint thinner to the viscosities given below. Varnished castings shall be wirebrushed before priming to insure removal of small rust spots, loose varnish, and other defects. Allow the primer to air dry at least 4 hr. before applying the second coat.

PAINTING—SECOND COAT: Apply, over the primer, a coat of light gray paint, thinned with paint thinner to the viscosities given below. Allow this coat to air dry at least 4 hr. before applying the top coat.

PAINTING—TOP COAT: Apply, over the second coat, a coat of dark gray paint thinned with paint thinner or benzine to the viscosities given below. Allow this coat to air dry at least 18 hr. before handling.

THINNING: Thin the paints with proper thinner to the viscosities given below, using the Demmler Viscosimeter.

Viscosity (Seconds)
Seconds—Demmler Viscosimeter

Brushing	Pressure Tank Spray	Suction Cup Spray
100-300	80-150	25-60
	Flowing or Dipping 15-20	

Where viscosity control of paints cannot be followed, add approximate quantities of thinner as follows:

Brushing—1 pint thinner to 1 gal. paint
Spraying—2 pints thinner to 1 gal. paint
Dipping—3 pints thinner to 1 gal. paint

Standardization of finishing materials permits standardizing production procedure

Establishing Standards for Materials—II

R. E. HELLMUND, *Chief Engineer*
D. F. MINER, *Manager*
Materials and Process Engineering Department
Westinghouse Electric & Manufacturing Company

In this second and concluding installment the authors define the classes of standards and explain how manuals covering general standards are prepared for inter-plant use. Part I appeared in the March number

IT IS impossible to bring all practice into line with perfect standards. Special varieties and sizes of materials will always be necessary. The recognition of this led to the adoption of "Class *A*" for general standards and "Class *B*" for restricted materials—that is, specials. Class *A* stocks are used freely and always in preference to Class *B* to bring about large quantity use of these materials and consequent reduction in cost. With reference to Class *B* stocks there are definite restrictions regarding their use and they should be resorted to only when satisfactory results cannot be obtained with Class *A* stocks. To make these restrictions effective, special approval is necessary for the use of Class *B* stocks either by the department head or the standards engineer. Class *B* stocks are identified, as their specifications are printed on a special color paper and definitely marked to indicate the restrictions under which they can be used.

There are several reasons for restrictions in the use of materials, as follows:

(a) Obsolescence: Over a long period of operation it is unavoidable that many materials will accumulate which do not fit in with a consistent plan of standardization. If such materials form part of existing or obsolete designs, they may be needed for repair parts. They should be carried as Class *B* stock but not to be used for any new designs and in this way they will gradually disappear as the old designs and parts are no longer produced.

(b) Special sizes for large activity: In large production, economy of fabrication will sometimes justify the use of a special size of material or one with tolerances for particular purposes. For example, where strip materials can be fed into forming machines it is frequently economical to obtain strip of a special width to avoid scrap or finishing operations. The same may be said of material with extra finish or rounded edges which will eliminate manufacturing operations. Merchandize, such as electric household appliances, refrigerators, or watthour meters, is made in sufficient quantities to warrant special materials for some parts. However, it is undesirable to introduce such special materials indiscriminately into other designs or departments and they should therefore be Class *B* items. This practice will facilitate their quick elimination when they are no longer needed for the particular purpose for which they were introduced.

(c) Special customer requirements: This is another instance requiring limitations in use. The U.S. Navy frequently specifies materials or finishes not usually employed for industrial purposes. Utilities sometimes have their own local standards for color of porcelain glaze or for hardware. Unless it is advisable to adopt these standards for general use, they should be kept out of the general standards, and, being limited-use standards, should be listed as Class *B* materials.

(d) Local plant conditions: These conditions involve special processing equipment or sources of supply. They may require materials not desired as standards for other plants. As examples: a humid location may require a special formula for lacquer thinner; a certain foundry sand may be used because of its proximity to the plant; or the type of baking oven used may require modification of insulating varnishes. These conditions may be absent at other plants and the use of such materials should be restricted or avoided at these other plants. This may again be accomplished by listing for local use, or as Class *B*, or by some other warning identification.

It is commendable to standardize the major materials within each plant of a large company. Local effort must be relied upon to make detailed decisions and to

make the work effective. A bigger task appears, however, when an attempt is made to coordinate the materials of fifteen plants manufacturing products as diverse as clock motors and turbines, or radios and oil circuit breakers. When an attempt is made to have all plants conform to a general plan on standards, the general excuse is that the problem is different in the several plants. This is true in detail, as the individual parts of any one plant are different, but there are points of similarity and interests in common.

Notwithstanding the difficulties involved in tying units of a company together, the goal is worth striving for and the reward is great. It would be foolish to insist on complete uniformity, as such a program would be impossible to carry out, but the major items in common at two or more plants can be correlated to a remarkable degree. The desired goal will never be reached in any company, but the aim should be to cover as many lines of activity as can be justified or where the rewards are the greatest. The value of this is appreciated when the time comes to move the manufacture of a product from one plant to another.

Coordination among the various plants with reference to standards in sizes is of course as desirable as coordination on grades. The stock of sizes kept at each of the plants may be a selection from headquarters sizes or may cover a different range or may be more comprehensive than the headquarters list. However, all sizes should be coordinated. This can be readily done by making plans for the rational selection of sizes which will embrace all possible requirements of all plants.

One procedure which helps the inter-plant problem is to recognize the dominance of certain plants in certain materials. A plant engaged in making household appliances is certain to be a leader in finishes and appearances. The standards of that plant with respect to such subjects as electroplating may well be copied by other plants. Another plant may build equipment subjected to high temperatures; materials such as heat and corrosion-resistant steels are perhaps best known by that plant. The final selection of standard materials at the dominant plant often can be made a company standard with some modifications. A plant making welded, liquid-containing vessels may well be looked to for the best welding materials and gaskets. In this manner something can be found in which each plant can be considered the leader. No plant wants to feel that it is dictated to by another plant or by headquarters, especially in an activity in which it has had more experience than the other plants.

Information on Standards

The benefits of standardization of materials cannot be obtained without provision for adequate information easily available to all who use the materials. One of the accepted methods of handling this problem is the distribution of books or manuals of standards. There is a real advantage in having all information of this character in one place if the volume is not too large.

The plan adopted by the company under discussion was to issue a "general standards book," containing over 300 pages, at the largest plant, which is also the location of headquarters of staff activities. This book is made up of three sections, the contents of each being

Portion of the hardware storeroom. Out of 1,350 former sizes of metal washers, only 150 were retained

Fewer grades and sizes of sheet steel means more advantageous purchasing and less stores expense, as well as lower inventories

Another source of saving. Different grades of materials sometimes require different shop equipment

arranged alphabetically. In the first section are listed the headquarters plant stock sizes of all generally used small parts, such as ball bearings, all forms of hardware, lubricating devices, and other units with detailed dimensions. The second part lists all stock raw materials, from asbestos to zinc. The third section, printed on pink paper, lists the Class *B* (non-standard) materials. Engineering departments, cost units and manufacturing units at the headquarters plant have one or more copies of the book, which is kept up-to-date by periodic revision. All other plants have one or more copies for reference. These copies are used as a guide to local standards and as an index of materials available if it is desirable to draw on headquarters stock. Furthermore, nearly one-third of the pages contain engineering data on properties, costs and other information, which is of use in all plants using the materials concerned. Each of the other plants has its own local standards book prepared on the same principles as the general book.

It has been found necessary to include many sheets of standard information on material properties. Although engineering handbooks are available, the designer needs to know the properties of the particular grades covered by the company's specifications. Comparison of grades, data on treatments, machinability, welding characteristics, and magnetic or dielectric properties are listed for many subjects and are usually useful to all units or plants.

Many of the benefits of material standardization are qualitative and therefore difficult to evaluate. However in many instances savings result that can be calculated and appreciated. Two typical examples show what has been done.

Metal washers occur in most every product assembly. Before any attention was given to this subject, each department designing its own washers and authorizing stock for their use, there was an astounding array of washers of all sizes. When a study was undertaken it was found that there were 1,350 varieties, and that several sizes had been designed as many as five times by different departments. When standardization had been completed, 150 sizes were found to be adequate. The saving in one year was $25,000 because of reduced inventory and lower tool-replacement costs.

As another example, over a period of years the number of special fabrics used for treated insulation had grown until the greatest activity was on materials that were not market standards. A study was undertaken to find out whether standard grades could be used. In the first year many items were changed to readily available goods subject to normal competition. The amount saved was $50,000. The next year other cloths and tapes were studied and an additional saving of $50,000 was made, resulting in a total of $100,000 a year on these items alone.

Consolidating Specifications

Simplification of the records and documents necessary for ordering and stocking materials is another important advantage. The use of consolidated purchasing specifications greatly reduces the number of papers to be handled and recorded and thus aids in reducing mistakes. Less clerical effort is required in maintaining the various forms of information necessary. As an example, the specifications covering a group of insulated fixture wires required 32 pages before consolidation. By eliminating the repetition of dimensions, ratings, and general instructions, and overlapping varieties, the consolidated specifications required only three pages.

The purchasing department is vitally interested in standardization of materials. Its problems in keeping down extra charges for small quantities, in eliminating errors from orders, and in obtaining materials on short notice are all made easier if the engineering departments follow a system of coordinated standards for materials.

Drives and controls, a major problem in the design of almost every machine, will be featured in the June number of *Product Engineering*. Latest developments in electric and hydraulic circuits, functional controls and drives will be covered in a series of major articles, reference book sheets and layouts of typical designs.

QUESTION AND COMMENT

Clamp-Ring Safety Brake

PHILLIP R. WHEELER
Alexandria, Va.

● In an effort to find a better one-way brake the clamp-ring type was selected for development. This patented clamp-ring type of one-way brake, or clutch, shown in Figs. 1 and 2, is simple in structure and positive in action. The large area of its gripping surfaces gives low unit pressures and a wear life that appears to be indefinite. The clamp-ring brake operates only when the shaft is stopped; or as a clutch when the shaft and clamp-ring have no relative motion between them.

It has been adapted particularly for use as an automatic safety brake, or back-stop device for use in automobiles. After considerable experimentation and testing, an installation was made in an automobile of light weight. This installation has been run more than 40,000 mi. with no apparent wear.

Reverse release, which has given difficulty in former devices, is in this instance inherently foolproof. The shift to reverse can be easily made at any time and under any condition of loading, regardless of tight quarters or of the steepness of the hill on which the car is to be backed down.

The general arrangement of the clamp-ring, used as an automatic safety brake, is shown in Fig. 1 and the schematic diagrams illustrated in Fig. 2 show the method of operation.

In Fig. 2a the shaft is turning forward, and the shear of the oil film between the shaft and the clamp-ring exerts a pressure on the lever arm at *A* and pins *B* and *C* in direction indicated by the arrows, thus opening the clamp-ring.

When the shaft comes to a stop, the oil film is partially squeezed out by the initial tension of the ring. The ring is provided with a single spiral groove to allow the oil to escape. Any tendency of the car to back causes a pressure at *A* and forces on the pins as indicated by the arrows in Fig. 2b. This locks the ring to the shaft.

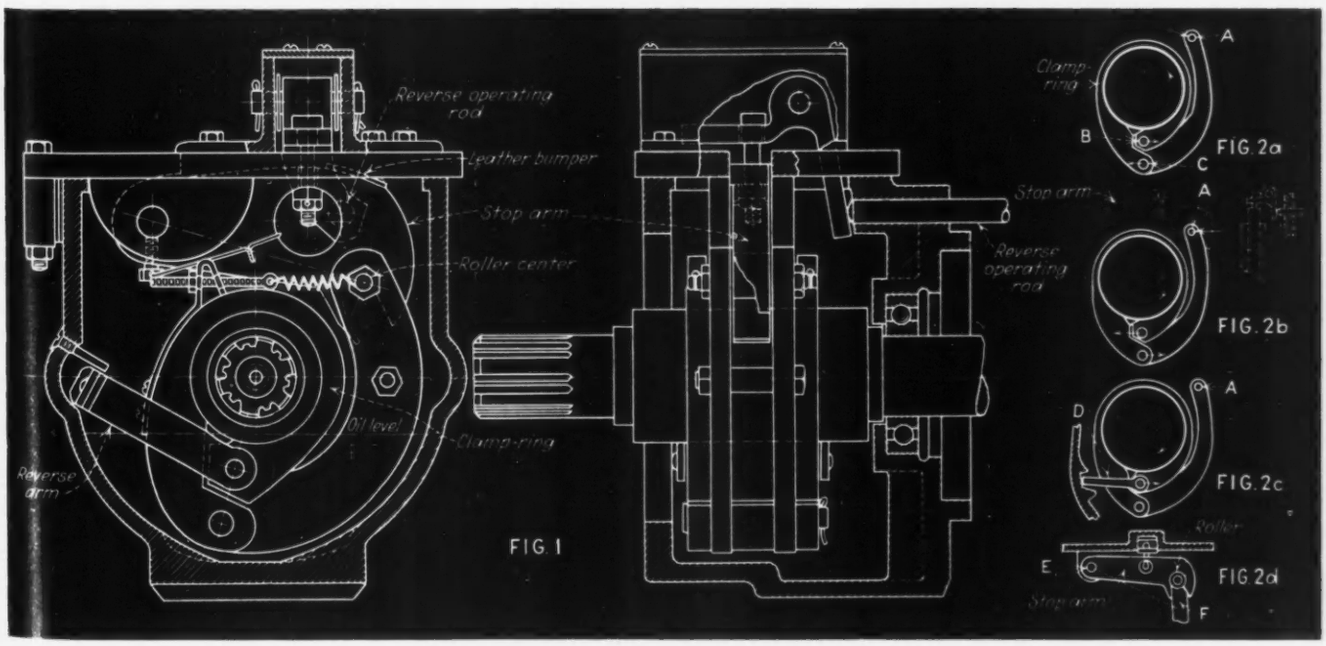
Shifting from neutral to reverse releases end *A* of the lever arm, and the clamp-ring then rotates about $\frac{1}{8}$ in. with the shaft, until the U-shaped member *D* comes in contact with the housing as shown in Fig. 2c. This holds the inside end of the ring, causing it to open as the shaft continues to turn backward.

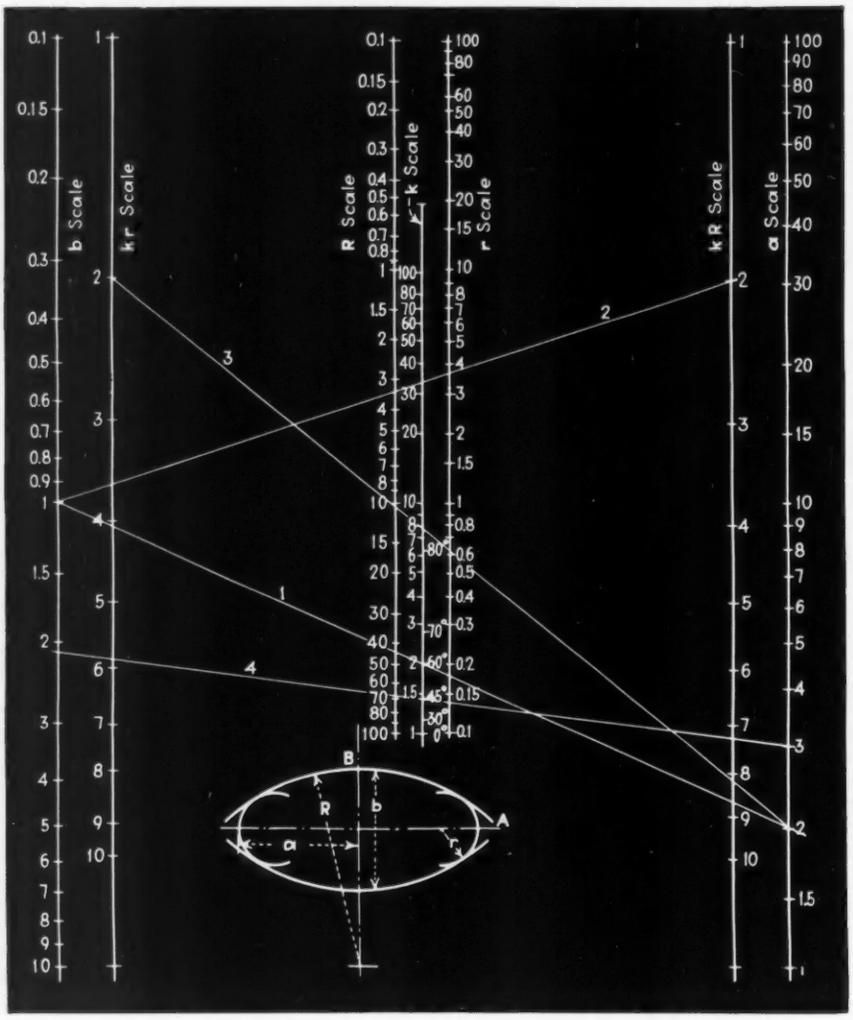
As shown in Fig. 2d, the stop arm is held in the housing by pin *E*. Surface *F*, which engages with the lever-arm roller, is cut on an arc the center of which is at pin *E*. When shifting to reverse the stop-arm is lifted, thereby releasing the roller and lever end. Regardless of the torque on the drive shaft, it is only necessary to overcome the friction of the roller to release the safety-brake.

A ratio of 3 to 1 between pins and point of application of force on the lever was found to be necessary for satisfactory operation. However, this ratio is not fixed as it depends upon various factors according to the installation. The position of or direction in which the lever should extend for optimum locking action was determined to be back around the shaft with the lever as near to the shaft as possible.

It was not necessary to take advantage of all the locking action which can be obtained by varying pin size or shaft diameter. Considerable less than the maximum obtainable will give positive locking action and keep shaft diameters, stresses and size of parts at a minimum. This particular installation on the light-weight car required only a 2-in. diameter ring.

Fig. 1—Automatic one-way brake used on light-weight car to prevent backing on a hill. Fig. 2—At Fig. 2a the shaft is turning forward, at Fig. 2b the shaft comes to a stop, at Fig. 2c is shown the reverse motion, and Fig. 2d shows the stop arm





Radii for approximate ellipses having eight points in common to true ellipses and practically the same true area

Alignment Chart for Approximate Ellipses

J. S. BEGGS
Research Laboratories, Eastman Kodak Company

● The accompanying chart gives radii for drawing approximate ellipses with compasses. These ellipses will have eight points in common with true ellipses and also will have practically the same area. The chart method is faster than the usual graphical method. The construction lines of the graphical method are usually difficult to erase on a complicated drawing. If the major and minor semi-axes of the ellipse are a and b respectively the chart gives the two radii R and r used in drawing the approximate ellipse as shown.

The use of the chart may be illustrated clearly with a concrete example. Assuming the major axis A to be 4 in. and the minor axis B to be 2 in., then $a = 2$ and $b = 1$. The procedure for obtaining radii R and r is as follows:

(1) Draw the line (1) from the point 2 on the a scale to the point 1 on the b scale. Where this line cuts the k scale read off the value of $k = a/b$ which is 2 in this instance.

(2) Transfer this value of k to the k R scale. From 2 on the k R scale draw the line (2) to the point 1 on the b scale. On the R scale is read the value $R = 3.28$ in. where the R scale is cut by line (2).

(3) Similarly transfer the value of k to the k r scale. From 2 on the k r scale, draw the line (3) to 2 on the a scale. Where the line (3) cuts the r scale is read the value $r = 0.610$ in. Checking these R and r values, $R\,r = a\,b$ or 2 in this instance.

If a circle having a 3-in. radius is to be projected on a plane at 45 deg. to the plane of the circle, its projection will be an ellipse with a 3-in. semi-major axis. To find b, the semi-minor axis, draw the line (4) from the point 3 on the a scale through the 45 deg.-point on the k scale to cut the b scale. This gives $b = 2.12$. The dimensions for R and r are obtained as explained in steps (1), (2) and (3) above.

What Shape of Tooth?

To the Editor:

● I want to use a positive tooth or dog clutch that can be engaged while rotating. It is to be used in a reversing installation where excess noise is objectionable. Can your readers give me some idea as to the shape of tooth that will give these characteristics? What materials and heat-treatments are best suited for medium loads and speeds and for satisfactory wear? The teeth are to be lubricated by a splash system.

—JOHN BEST
Baltimore, Md.

Pinion Centers for Planetary Gears

To the Editor:

● The author of the article with the above title in the January issue of *Product Engineering* has evidently lost sight of the true action of planetary gearing for, though his explanation and derivation of the formula are quite involved, an error in his analysis has produced a formula which gives a false result. Rather than try to point out where the author introduced his error it seems more practical to attack this problem from the beginning and derive a formula by the most simple method.

Assume that the gears are assembled as in Fig. 2 on page 24 of the January issue, with the line of centers of the sun and planet gears passing through the center of a tooth on the sun gear and the center of a tooth space on the ring gear. With the sun gear held stationary, rotate the ring gear one tooth (1/56 of a turn). Now the center of the second tooth space of the ring gear lies on the original line of centers and bears the same relationship to the sun gear tooth as the first tooth space on the ring gear did before it was advanced one tooth. Therefore, disregarding face widths of the gears, another planet gear can be assembled in the position formerly occupied by the first planet gear and the line of centers of these gears coincides with the original line of centers. The angular distance through which the first planet gear rotated while the ring gear was rotated one tooth, which is the angular distance between the two planet gears, is the minimum distance between meshing planet gears.

In a planetary gear train with a stationary sun gear the planet gear makes $N_r/(N_r + N_s)$ revolutions around the sun gear while the ring gear makes one revolution. Therefore, when the

ring gear makes $1/N_r$ revolutions (one tooth) the planet gear makes $1/N_r[N_r/(N_r + N_s)]$ revolutions around the sun gear or $1/(N_r + N_s)$ revolutions. Expressed in degrees, the angle of rotation $\theta = 360/(N_r + N_s)$ which is the minimum distance between meshing planet gears.

Applying these formulas to the example in Fig. 2, while the ring gear makes 1/56 of a revolution (one tooth) the planet gear makes $1/(56 + 18)$ or 1/74 of a revolution around the sun gear and $\theta = 360/74 = 4$ deg. 51 min. 54 sec. The point nearest to 120 deg. is 25/74 of a circle or $25 \times 360/74 = 121$ deg. 37 min. 18 sec. Applying the formula to the example in Fig. 1, the planet gears will mesh 360/72 or 5 deg. apart instead of 10 deg. as stated in the original article. Stated simply, the number of meshing positions is equal to the total number of teeth $N_r + N_s$ in the ring and sun gears. —C. A. JOHNSON
Springfield, Mass.

To the Editor:

● Mr. Johnson's method and solution are correct. A small checking model was made of 36 diametral pitch gears as illustrated in Fig. 2 on page 24 of the January issue. The 18-tooth sun gear meshes with three 19-tooth planetary gears, the latter meshing with a 56-tooth ring gear. It was found that 74 meshing points occur. Therefore the number of meshing points are equal to the sum of the teeth in the 18-tooth sun gear and the 56-tooth ring gear. The angular setting nearest 120 deg. in Figs. 1 and 2 as in Mr. Johnson's solution are also in agreement with the tests made. —W. SPENCER
Denver, Colo.

Wrong Credit

To the Editor:

● In the design shown as Fig. 9 in page 31 of your January number it is stated that this coupling is manufactured by Foote Gear Works. As far as I know, this company does not make flexible couplings, and the design shown is a product of the Ajax Flexible Coupling Company. Undoubtedly, if the Ajax people notice this mistake, they will call your attention to it. Similarly, the design shown in Fig. 11 is not a product of Foote Gear Works, although I do not know whose design it is. —G. I. W.
Baltimore, Md.

EDITOR'S NOTE: It will interest G. I. W. to know that W. I. Belden, sales manager of the Ajax Flexible Coupling Company, of Westfield, N. Y., beat him to it by a week. We must humbly admit the mistake. The design shown in Fig. 9 is a product of Ajax and the design in Fig. 11 is a former product of Foote Bros. Gear & Machine Company. It is true that Foote Gear Works, whose principal product is gears and reduction units, do not make flexible couplings. Foote Bros. Gear & Machine Company makes a cushion type coupling as shown in Fig. 10 with the same pin and bushing construction.

Babbitt for Sleeve Bearings

To the Editor:

● In 1839, the U. S. Patent Office granted an application from a silversmith for a patent on something entirely unrelated to his work—a locomotive bearing-box. The claims on which the patent was granted were that by lining the boxes "within which the gudgeons or journals of machinery in general and the axles of railroad cars, locomotive engines, and other cars or carriages are to run" with a metal composition such as "hard pewter" the "heating and abrasion which are apt to occur under the ordinary mode of constructing them are prevented, and their durability is consequently increased." This alloy contained approximately 90 per cent of tin, the balance antimony and copper. The inventor's name was Isaac Babbitt.

Since Isaac Babbitt's invention, many alloys have been granted patents on claims of superiority over the original alloy but very few of the patented alloys have proven successful as bearing metals. No machine today in general use has caught up with the capabilities of the modified britannia metal. It still is capable of carrying the load and withstanding the shocks and the speed of any machine now in general use.

Neither the automobile, nor the diesel engine, nor the rock crusher, to mention three machines in which the bearings are subjected to very severe service, needs a better bearing alloy than the genuine babbitts that have been used for nearly a hundred years. In automobile engines the newer alloys might wear longer than babbitt, but a babbitt bearing will, under normal conditions, serve for the life of the car. In the Indianapolis automobile races of the past several years, where the connecting-rod bearings were subjected to speeds as high as 6,000 r.p.m. the old fashioned babbitt came through in splendid shape.

One important advantage of babbitt over most of the new alloys is that a bearing can be lined in any machine-shop. Most of the new bearing alloys require expensive equipment, or special skill, or both. Until a new alloy is developed that can be handled as easily and as inexpensively as babbitt, and possesses better bearing and wearing properties, and until the machinery in which bearings play an important part shall require that they stand up under much heavier loads, much harder knocks and much higher speeds than present-day machinery, babbitt metal will continue in its present position of supremacy. —HENRY MARTIN
New York, N. Y.

To the Editor:

● I am quite certain that Mr. Martin is misinformed in some of his statements, particularly where he states that tin babbitts are quite satisfactory for automotive uses. I can assure you that if this is true, the Ford Motor Company is wasting hundreds of thousands of dollars per year on their bearings and that there are many installations of diesel type automotive engines where tin-babbitt bearings will not stand up long enough to give a satisfactory test of the motor. It is quite true that automobiles can be designed so that tin bearings will be satisfactory, but at the present time several of the automobiles as they are now designed, cannot use tin-babbitt bearings.
—A. J. PHILLIPS
Maurer, N. J.

Rubber Under Pressure

To the Editor:

● When pressure is placed on a block or cylinder of rubber the dimension in the pressure direction is reduced and other directional dimensions increase. This principle is used to advantage in the design of various types of holding devices, and should be adaptable for a number of other applications. Such an installation is that of rubber sanding drums of small size, designed for sanding the scrolled edges of various types of work.

Use of this type of drum allows quick changing of the sandpaper or other coated abrasive, which is made up in the form of sleeves, in such manner as to slip readily over the rubber drum. The drum is simply a small cylinder of rubber centrally bored to slip over a revolving arbor and held by the arbor washer and nut. The drum, with the coated abrasive sleeve placed on it, is slipped onto the arbor, and the arbor nut tightened. As the nut presses the washer against the end of the rubber cylinder, the rubber is compressed endwise, and increases sufficiently circumferentially by the same action to hold the abrasive sleeve firmly. It is obvious that not only holding action, but braking and other applications might easily be based on this principle. —JOHN E. HYLER
Peoria, Ill.

Typical Designs of

CYRIL DONALDSON
Rochester Mechanics Institute

This last installment shows designs wherein keys, gears, chains, rollers, cushions or springs are used for the flexible member

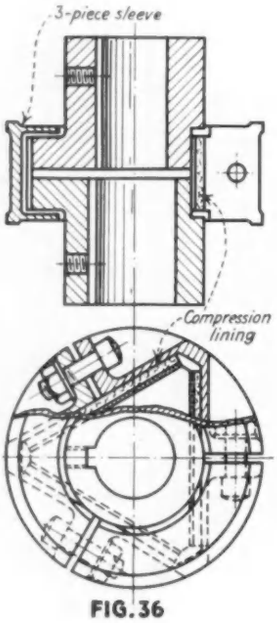


FIG. 36

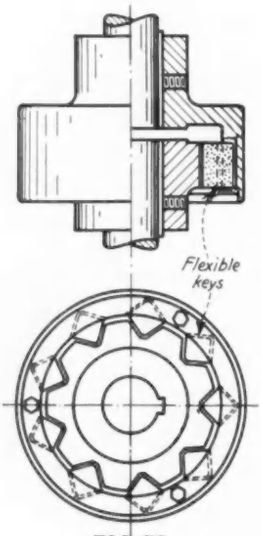


FIG. 37

Fig. 36—In this coupling of Falls Clutch & Machinery Company two hubs with triangular heads and a three-piece sleeve are used. The sleeve is bolted together when assembling. Three pieces of compression lining provide the necessary flexibility. Misalignment is compensated for in all directions by compression of the linings.

Fig. 37—This Medart Company flexible coupling uses a number of square keys or pins made of fiber, Textolite or other flexible material which engage V-slots. Clearance is provided in the V-slots for flexibility. The pins are held in place by a retaining collar. Coupling can float endwise.

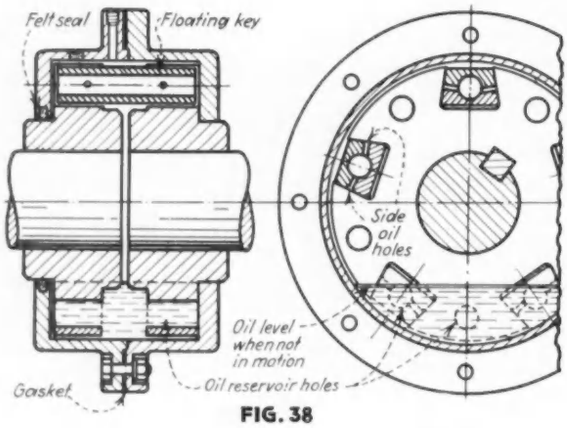


FIG. 38

Fig. 38—In the design of The W. H. Nicholson & Company flexible coupling, a series of floating steel keys slide in dovetail slots cut into each flange. The degree of misalignment compensated for depends on the clearance between the keys and slots. Wear is reduced and cushioning is provided by operating keys in oil bath. Keys act noiselessly, centrifugal force keeping them against the slot surfaces.

Fig. 39—In another design made by Falls Clutch & Machinery Company, a triangular center floating member made of steel is placed inside two flanges. As in Fig. 36, three pieces of compression lining are used. Coupling flanges are triangularly recessed.

Fig. 40—In this Clark Controller Company design, a splined hub mounted on each shaft is connected by a sleeve having internal projections. Power is transmitted through strips of packing fitted between the projecting teeth in the hubs and sleeve. Packing is retained at each end by keeper ring and snap ring. Compensates for misalignment in all directions without the use of flexing materials.

Fig. 41—In one design of Lovejoy Tool Works flexible coupling, individual free-floating load cushions are hung

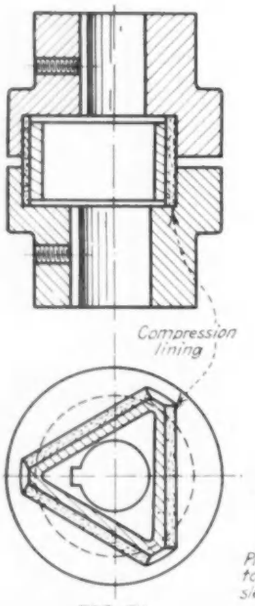


FIG. 39

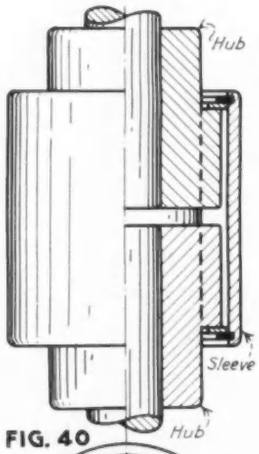
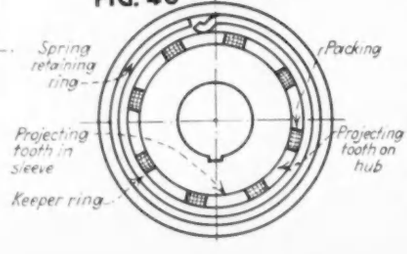


FIG. 40



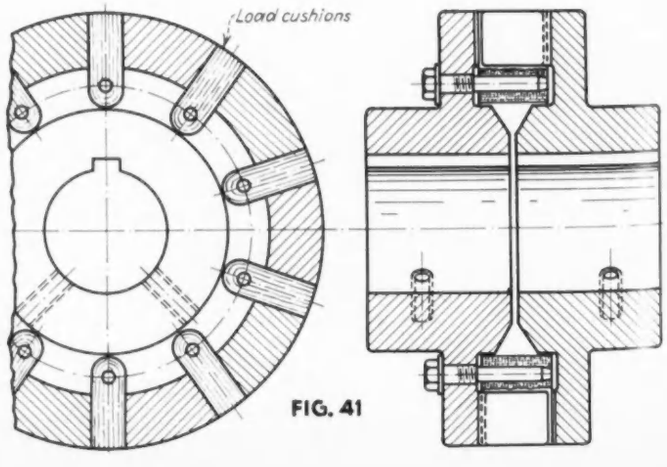


FIG. 41

Flexible Couplings–IV

between the flange jaws on removable studs. These replaceable cushions are made of brake-lining material, leather or rubber-duck fabric, depending on the loads sustained and the resilience required. No lubrication is needed.

Fig. 42—The positive drive design of Diamond Chain & Mfg. Company consists of two sprockets connected by a length of roller chain. Clearance between sprockets and chain side plates allows freedom to compensate for misalignment in all directions.

Fig. 43—The Poole Engineering & Machine Company uses a two-piece floating sleeve with internal gear teeth cut at each end, meshing with gear teeth on hubs. Toothed hubs are mounted at the end of each shaft. The hub teeth have spherically formed crowns. The teeth are in mesh around their entire circumference. Compensates for misalignment in all directions without the use of flexing materials. Bearing surfaces are lubricated in a bath of oil. Dust is excluded by packing ring at either end.

Fig. 44—A silent chain is used as the flexible member in the Morse Chain Company coupling, the load being distributed over a number of teeth. A series of retaining links, running in the center of one sprocket, keep the chain in place. Flange covers enclose the chain when necessary.

Fig. 45—Convex jaw surfaces that exert a rolling pressure when loaded are used in another Lovejoy Tool Works coupling design. The convex surfaces are so proportioned that the compression is uniform over the entire area of each spider arm. The floating spider is made of a resilient material which gives flexibility in all directions.

Fig. 46—This T. L. Smith Company type of coupling has a flexible metal ring engaging projections integral with the outer and inner hubs. A packing ring protects the interior from dirt, yet compensates for angular misalignment. Coupling can drive in either direction.

Fig. 47—In this Meriam Company design, the internal and external hub is connected by a series of spring steel rollers fitted into semi-circular recesses in each hub. The rollers are made of strip steel, wound spirally and ground on the periphery. Quiet operation is secured by packing the interior of the coupling with grease or ground cork.

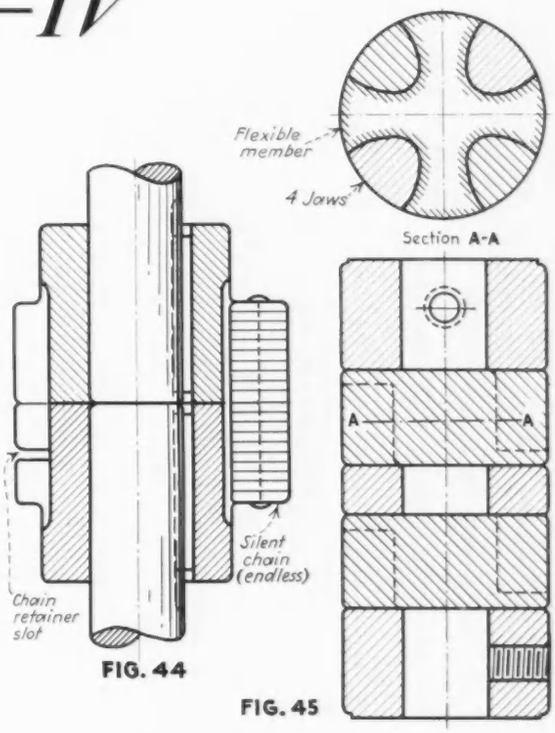


FIG. 44

FIG. 45

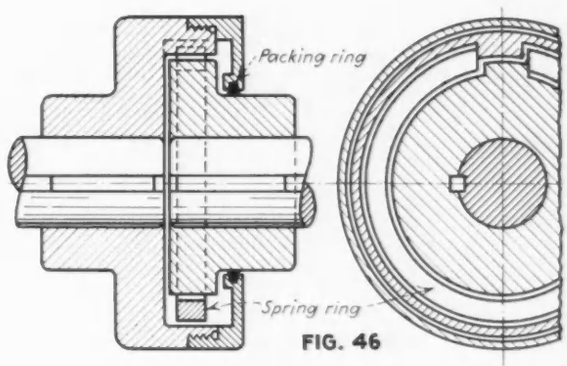


FIG. 46

FIG. 47

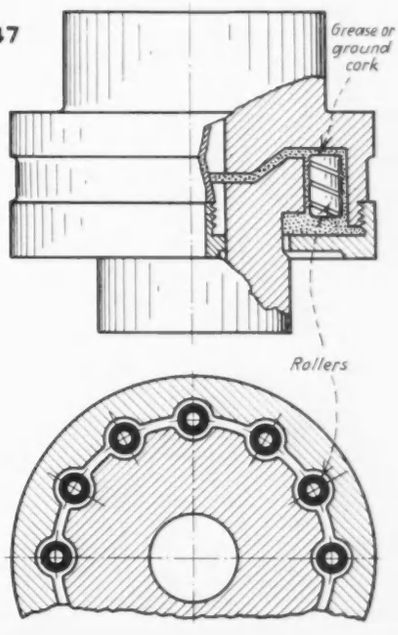


FIG. 42

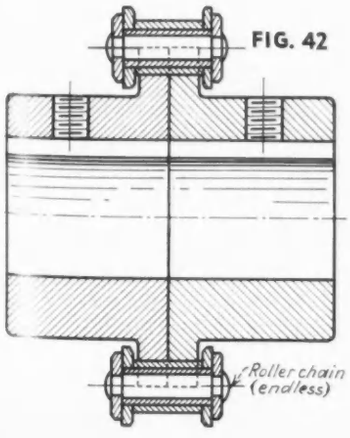
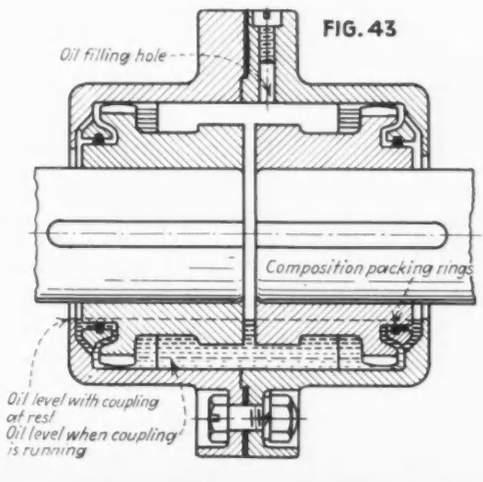


FIG. 43



NEWS

A.I.E.E. Celebrates 50th Anniversary of First A.C. System

IN A nation-wide celebration sponsored by the American Institute of Electrical Engineers, simultaneous meetings in 48 principal cities throughout the United States celebrated on March 20, the 50th Anniversary of the first successful demonstration of the first alternating current system. On March 20, 1886, at Great Barrington, Mass., William Stanley completed the first a.c. electric system, furnishing light to 13 stores, two hotels, two doctor's offices, 1 barber shop and the telephone and post offices. The lamps used were 150, 50 and 16 candle power sizes.

Interesting facts about this first installation is that the 25 hp. steam engine drove the generator which furnished 12 amp. of current at a maximum of 500 volts. The voltage was then stepped up, transmitted over the line and the voltage stepped down in transformers housed in wooden boxes in the cellars of the buildings to be lighted.

Following this initial demonstration there was a period of intense production activity with results at an amazing speed. By September of the same year designs were completed, machines built, tested, and put into practical operation. In November George Westinghouse, who was associated with Stanley, installed the first commercial system in Buffalo, N. Y. By October, 1877, between 30 and 40 plants were in successful operation generating and distributing a.c. current.

It was George Westinghouse who, through his vision and commercial enterprize, and William Stanley, who through his genius and consistent effort under conditions of extraordinary difficulty, made this development possible.

Stanley was an intensely practical technical man. If something did not work one way he tried other ways and kept experimenting until he obtained his goal. In 1888 he was joined by C. C. Chesney, later a vice-president of the General Electric Company, and in 1889 John F. Kelley, electrician of the Newark Shops of the Westinghouse Electric & Manufacturing Company became associated with Stanley. These three men combined their efforts for the continued development of a.c. transmission. Not until 1891, when Steinmetz met Stanley, did the mathematical analysis which became the basis for the design of alternating current machinery, come into the picture.

Start Accrediting Engineering Colleges

THE PROGRAM of inspection and accrediting of engineering curricula, which the Engineers' Council for Professional Development offered last year to schools granting engineering degrees, has been accorded a hearty response by 34 colleges and universities in the New England and the Middle Atlantic States. Inspections in these two regions were inaugurated late in 1935 and will be actively continued during the next few months. When substantial progress has been made in these areas, it is expected that the program will be extended to engineering schools throughout the United States.

The accrediting program has for its purpose the best development of engineering education by identifying those institutions that offer engineering curricula worthy of recognition as such. The second objective is to build up a list of accredited engineering schools, which it is hoped may be uniformly adopted by educational, technical and State organizations now using dissimilar lists.

Institutions in the New England states that have applied for accrediting are: Brown University, Dartmouth College (Thayer School of Civil Engineering), Massachusetts Institute of Technology, Northeastern University, Norwich University, Rhode Island State College, Tufts College of Engineering, University of Maine, University of New Hampshire, University of Vermont, Worcester Polytechnic Institute, Yale University.

In the Middle Atlantic states the following schools have applied: Alfred University, Bucknell University, Carnegie Institute of Technology, Clarkson College of Technology, College of the City of New York, Columbia University, Drexel Institute, Johns Hopkins University, Lafayette College, Manhattan College, New York University, Newark College of Engineering, University of Pittsburgh, Polytechnic Institute of Brooklyn, Princeton University, Rensselaer Polytechnic Institute, Rutgers University, Stevens Institute of Technology, Swarthmore College, Syracuse University, Union College, Webb Institute of Naval Architecture.

In this activity, being advanced by the Committee on Engineering Schools headed by Dr. Karl T. Compton, president, Massachusetts Institute of Technology, ECPD is acting as an accrediting agency for seven technical, educational and legislative bodies. Other major committees of ECPD are giving their attention to selection and guidance of prospective engineering students, professional training of young engineering graduates, and the development of standards of professional recognition. ECPD headquarters at 29 West 39th Street, New York, N. Y., will supply further information.

Beryllium Copper Molds Open New Uses for Plastics

IN THE August 1935 number of *Product Engineering*, mention was made of the development of cast beryllium-copper dies for making molded plastics. This development, the result of two years of carefully planned experimental work carried on by the Bronze Division of the Gorham Company has now reached a point where its scale of use and limitations has been more definitely established.

It is pointed out that beryllium copper should be considered supplementary to the mold materials now employed and is not to be considered as a panacea or cure-all. It is a new tool and like all new tools it will be some time before its exact place in the industry will be determined. At present, cast beryllium copper molds have been found most suitable and economical for small parts of intricate designs that would need a milling cutter so fragile that it would not stand up in cutting the cavity in mild machine steel. Another definite field for the

beryllium-copper cast mold is for reproducing decorative ornamental designs where reproduction is obtainable in steel only by expensive hand engraving.

The characteristics of beryllium copper which make it particularly suitable for this purpose are the sharp definition that is obtained when the metal is cast and the ease of hardening the metal by a heat-treatment that does not warp the die or change its dimensions. Of course the dies do not last as long as those made of tool steel or pack hardened steel.

Alloy Steel Reduces Inertia Loads

A PENNSYLVANIA R.R. locomotive equipped in July, 1934, with Timken roller bearings for main and side rods recently completed 96,000 mi. in regular passenger service between Pittsburgh and Columbus, Ohio, without any trouble or defects having developed in the bearings or related parts.

In the application of these bearings, the crankpins, crossheads, piston rods and heads were completely redesigned, a nickel-chromium steel corresponding to S.A.E. 4340 being used for these parts. Heat-treated, this material developed a yield point of 120,000 lb. per sq.in., and made possible a weight reduction of 240 lb. per main crankpin and 508 lb. in the weight of the reciprocating parts, thereby reducing the crankpin loads approximately 48,500 lb. at 100 mi. per hr. speed.

MEETINGS

Society of Automotive Engineers—National Tractor and Industrial Power Equipment Meeting, Milwaukee, Wis., April 15-16. John A. C. Warner, secretary, 29 West 39th St., New York, N. Y.

American Gear Manufacturers Association—Annual meeting, Adelphia Hotel, Philadelphia, Pa., April 20-21. J. C. McQuiston, secretary, Penn-Lincoln Hotel, Wilkinsburg, Pa.

Society of Automotive Engineers—Summer meeting, White Sulphur Springs, West Virginia, May 31-June 5. John A. C. Warner, secretary, 29 West 39th St., New York, N. Y.

EXHIBITIONS

National Oil Burner Show—Convention Hall, Detroit, Mich. April 14-18. C. F. Curtin, 30 Rockefeller Plaza, New York, N. Y.

Midwest Power Engineering Conference and Midwest Engineering and Power Exposition — International Amphitheatre, Chicago, Ill., April 20-24. G. E. Pfisterer, manager, 308 W. Washington St., Chicago, Ill.

New Standard for Pipe Thicknesses

AN American Tentative Standard for Wrought-Iron and Wrought-Steel Pipe (B36.10-1935) was recently approved by the American Standards Association. It was developed by the ASA, under the joint leadership of the American Society of Mechanical Engineers and the American Society for Testing Materials.

Copies of the new standard are available from the American Society of Mechanical Engineers, 29 West 39th Street, New York, or from the American Standards Association, at 50 cents each.

Die-Castings Exhibit Held in New York

WHAT is claimed to be the most comprehensive showing of die castings ever held in this country is now on exhibit at Metal Products Exhibits, Inc., International Building, Rockefeller Center, New York City. It will continue until May 2, according to the announcement of the American Die Casting Institute.

Twenty-seven commercial die-casters are cooperating to make this exhibit the most comprehensive showing of die castings ever held in this country. In many of the exhibits enlarged production photographs, the actual die castings and the finished products into which they are assembled are displayed to tell a complete production story.

Many assembled products newly designed for die castings are shown, including vacuum sweepers, industrial machinery, decorative novelties, home workshop tools and toys. These indicate the range of applications and suggest that the exhibit will cover the high spots of many industrial shows.

Do You Know That—

• Materials such as copper and aluminum having high electrical conductivity now can be resistance welded by the use of a timing device that limits duration of current to 0.001 sec. or less. (1)

• Effective strength of spot welds is greatly increased by using two parallel rows of spots. (2)

• In testing the transverse strength of cast iron, using the so-called arbitration bar, it was found that the strength increased with increase of the maximum temperature to which the cast iron had been heated. (3)

• Cast stoker links of 15 per cent chromium cast iron resist growth even when repeatedly heated above a red heat. (4)

• Stainless or chrome-moly steel is used for a new design of light-weight anchor for yachts and boats. The fluke is designed for easy penetration in the hardest bottom and holding power is claimed to be 25 times that of a conventional anchor of equal weight. Corrosion resistance and light weight are advantages for evident reasons. (5)

• In salt manufacturing plants even the hand shovels are made of Monel to resist corrosion. Nickel clad steel and nickel alloy cast iron are also used extensively. (6)

• The average weight of die-castings per car in 1935 models was 25 lb. while one model used as much as 80 lb. per car. Most of these die-castings were zinc base alloys. (7)

• A new bicycle recently put on the market has no pedals, chains or gears. Yet it can be propelled by the operator merely shifting his weight, which can be done without movement of the feet. The explanation lies in the rear wheel being mounted with its axle eccentric. (8)

• Fibrous spun glass woven into blankets is one of the best heat insulating materials. (9)

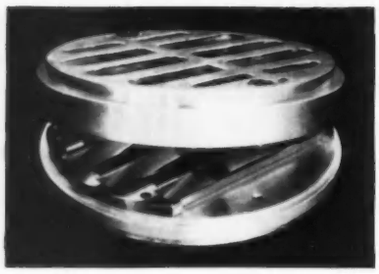
Great Lakes Exposition

THE steel and iron industry and manufacturers of heavy machinery will play an important part in the Great Lakes Exposition, to be held on the Cleveland lakefront June 27 to October 4.

The exposition celebrating the centennial year of Cleveland is to be staged on 125 acres of downtown Cleveland lakefront. Exhibits are being prepared by manufacturers and business concerns throughout the entire great lakes region.

New Materials and Parts

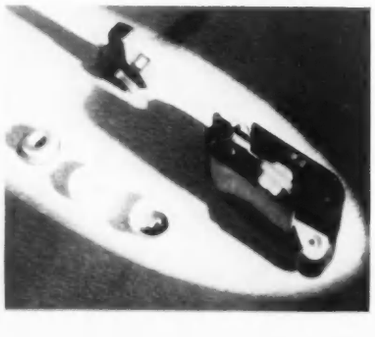
Channel Valve



This new type of valve is used in air and gas compressors. A greater portion of the valve opening is utilized which gives slower air speeds through the valve ports. The only moving parts of the valve are a number of "valve channels" within each of which is a flat spring, fitting within the channels so that when the valves lift the springs straighten and a small quantity of air is trapped between the springs and the channels. This construction is said to form air cushions, the function of which is to decelerate the valve channels after opening and bring them to a stop softly and quietly without impact. Because of the total elimination of impact, the valves are said to last longer than heretofore. The valve channels are made light in weight and because of their shape give strength and rigidity. The channels are said to lift straight off the seat without flexing, returning to the same position. The springs act only as springs and not as valves. The channels and springs are made of stainless steel to resist corrosion. Ingersoll-Rand Co., Phillipsburg, N. J.

Silent Mercury Switch

A small, compact electric switch, said to be completely silent in operation and to have no moving parts to wear out. The switch consists of two shallow chrome-steel cups about $\frac{3}{4}$ in. diameter, sealed together with a strip of lead glass to form a hollow compartment. Separating the cups is a disk of ceramic material in which there is a small hole located near the edge. The compartment is evacuated after fabrication into which approximately 4 grams of mercury are inserted. The mercury fills approximately one quarter of the space. After the insertion of the mercury, the compartment is filled with hydrogen at about atmospheric pressure, and sealed by welding. Hydrogen is used because of its cooling properties and because in it an arc can be better quenched. In the "off" position, the hole in the ceramic insulating disk is above the line of the inclosed mercury. When rotated 20 deg. the switch assumes a position where the mercury can flow through the hole, thus establishing a contact between the two chrome-steel cups and closing the circuit. The switch is cadmium plated. General Electric Co., Schenectady, N. Y.



Mercury Vapor Tube Printer

For exposing blue, brown or direct process prints. Model 7 is of the horizontal continuous type and accommodates tracings to 42 in. in width. The machine is made in upright console style with pedestal type base to give operator maximum comfort. Tracings are fed on an endless fabric belt contacting a curved glass. Uniform illumination is provided by an arrangement of $3\frac{1}{2}$ amp. 50 in. mercury tubes mounted in close proximity to the contact glass. Each tube is interchangeably and independently mounted. Can be supplied with either three or four tubes depending on intensity and speed desired. The use of three tubes permits a printing speed of 24 linear in. per min., and 30 in. per min. with four tubes using good tracings and fast blueprint or direct process papers. Variable-speed motor is controlled by a 28-point rheostat. Two-step motor pulley is provided for higher or lower speed ratio. Runs on a.c. or d.c., 110 volts, 20 amp. capacity. Can also be furnished for other voltages. The C. F. Pease Co., 813 North Franklin St., Chicago, Ill.

Vacu-Break Safety Switch

In the design of this switch the conventional procedure is reversed by placing the arc-extinguishing structure on the moving contact instead of on the stationary contact. The arc extinguisher is made in the form of a partitioned chamber, totally inclosed except for two slots at the bottom to receive the dual stationary contact in separate cells. The moving contact is floatingly supported in this chamber which moves with the moving contact in and out of the stationary contacts, as guides. The piston-action of the contacts when rupturing in this close-fitting arcing chamber are said to evacuate the small amount of air present in such a restricted inclosure, thus minimizing burning of the arc chamber or the contacts, by reducing the oxygen available to support combustion. By confining the arc in its own arcing chamber and then within the cabinet walls, it is claimed to be doubly sealed and can be operated under extreme overload with safety to the operator. The wire grips are in the form of solderless lugs. The current-carry-

ing parts are mounted on a molded base. A quick-make and break compression spring accelerates operation although the switch can be operated if the spring were broken or removed. The switch is made in three types; namely Master, Standard, and Junior types. It is claimed that the switches will withstand 100 per cent overload. The switches are housed in "Stylined" cabinets with handles to match. Bulldog Electric Products Co., Detroit, Mich.

Thermal-Induction Relay

For use in oil-immersed motor controllers. Has tripping characteristics such that difficulty in starting high inertia loads at full voltage is eliminated. A series line-current coil produces a flux in a magnetic core, and induces a current in a copper sleeve about this core. Heat is conducted and radiated to a bimetal strip which causes a set of contacts to open or close. The contacts, normally closed, have a slow opening action, preventing too quick tripping on overloads. Carbonization with resulting changes in the characteristics of the relay and sludging of the oil are avoided. General Electric Co., Schenectady, N. Y.

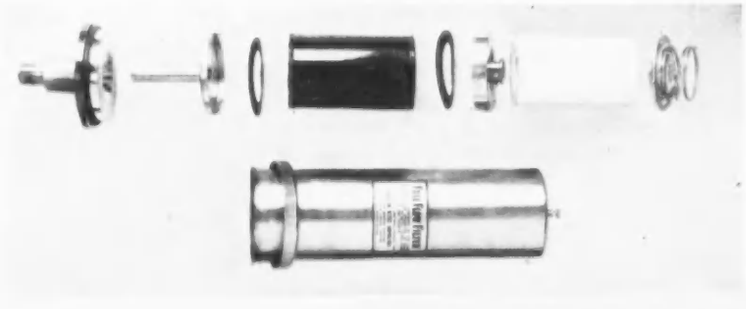
Remote Valve Control

A hand controlled valve for non-critical flows which need only infrequent readjustment. The valve can be mounted on a centrally-located panel from which the operator can adjust manually the degree of opening of an

inaccessible valve without moving from his station. The valve can also be reset to control temperature, pressure, flow or liquid level. It is said that the valve opening can be altered by as little as a few thousandths of an inch to produce the exact desired flow. There is said to be no pumping or vibrating and that the air pressure on the diaphragm is held exactly at the desired setting. The case is finished in lacquered cast bronze trimmed with buff nickel. The Foxboro Co., Foxboro, Mass.

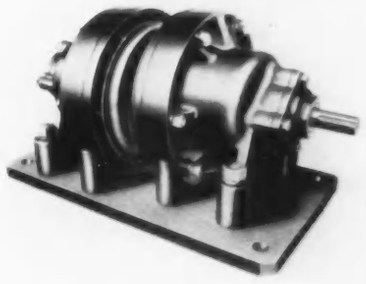
Full Flow Filter

Has combination honeycomb tube and carbon cartridge. The tube is of bleached cotton yarn wound on a wire mesh cylinder. This tube removes all suspended matter from the liquid being filtered. The cartridge made of highly-activated carbon receives the liquid from the honeycomb tube. The cartridge removes by absorption, for instance from water, chlorine and phenol and other objectional tastes and odors. Both the filtering elements are renewable, the tube being replaced every 30 days and the cartridge every 120 days. This time may be less, depending on the type of liquid being filtered. Made of brass, white nickel-dipped and is tested for 150 lb. pressure. Has a capacity of 1 gal. per min. Larger models to 60 gal. per min. are available. Commercial Filters Corp., 89 Broad St., Boston, Mass.



Mechanical Vibrator

Consists essentially of two offset weights mounted on shafts, geared together so that when one shaft is rotating the other must rotate at the same speed but in the opposite direction. This is said to permit directional control of the impulses or vibration in which the amplitude of vibration is controlled by the amount of offset weight. The size and speed of the unit, and the housing and bearings are designed to suit each specific application. Anti-friction bearings and heat-treated alloy steels are used in the construction and the housing is said to be oil-tight and dust-proof. The unit illustrated runs at 4,000 r.p.m. and delivers 8,000 strokes or blows per minute, each having a force of approximately 2,400 lb. The unit is made use of for conveying, screening and separating operations and other special applications. Ajax Flexible Coupling Co., Westfield, N. Y.

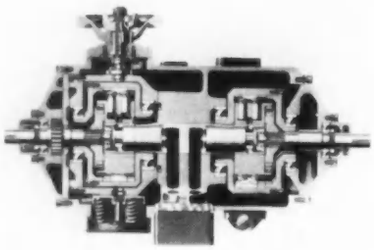


"No-Ink" Drawing Board Top

Is used for the elimination of inking of drawings. These drawing board tops are glued to an ordinary wooden board. It is of a white composition which is said to give a permanent white, smooth, drawing surface and to be tough, resilient, stain-proof, scratch-proof, mar-proof and washable with soap and water. No manila paper is used on this surface to back up the drawing paper. Regular drawing paper is attached directly to the surface, and a 3-H or harder pencil is used. All lines are drawn with a double stroke and with a slight pressure on the pencil. The pressure of the first stroke ridges the paper, and the back stroke fills the indentation evenly with lead. The result is a solid, opaque, even-edged line which is said to make blueprints and van dykes as clear-cut and readable as an ink line. It is said that changes on the drawings can be

readily made by the usual erasing process, and that the ridges or valleys will not show on reproduction. Because of the resiliency of the board, the indentations in the surface of the board top, made by the pressure of the drawing tools disappear immediately. It is also claimed that these drawings can be perfectly reproduced at any time as the lines will not smudge as the hard lead lies in the indentations. Sheets are $\frac{1}{8}$ in. thick and are available in any size. W. H. Long Co., 420 N. Clark St., Chicago, Ill.

Variable Speed Transmission



The Oilgear fluid power variable-speed transmission has been redesigned so as to simplify the input and output unit by eliminating 13 parts from each piston, permitting increased capacities and reduction in size. The friction and resistance on each piston has been lessened. It is claimed that the output end exerts 88 per cent torque efficiency at starting, and 97.6 per cent at 120 r.p.m. The output speeds are infinitely adjustable from 5 to 1,090 r.p.m. These new transmissions are available in sizes from 2 to 100 hp. They are controllable either by hand or by automatic direct and remote devices. The Oilgear Co., 1319 West Bruce St., Milwaukee, Wis.

V-Belts

Two styles of "Multi-V" belts are available one is of single strand construction as shown at the left. This belt is especially designed to withstand flexing and stretching on small pulley, short center, high-speed drives. For larger drives, a stronger belt of laminated construction is available as shown at the right, which has a greater number of cords and several layers of stiffening fabric.

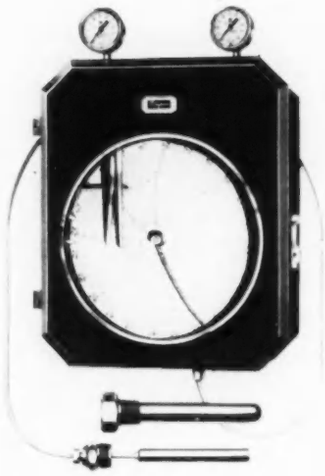
The "Long-life" V-belt for single drive has a fabric filler which is said to give transverse strength without loss of flexibility, and that the rubber impregnated filler cords resist stretch and provide longitudinal wear. The B. F. Goodrich Co., Akron, Ohio.

Acid Resisting Plastic Materials

A light-weight molded material known as 77 S B Black is used where acid resistance, frictional wear resistance and low water absorption are required. This Durez molding material has a weight of 20.6 grams per cu.in., or a specific gravity of 1.26 and a water absorption rate of 0.2 per cent A.S.T.M. rating. The material is also said to resist all the common concentrations of acids. It can be machined, sanded and buffed after molding, without exposing spots of filler or impairing the acid or friction-resisting qualities. General Plastics, Inc., N. Tonawanda, N. Y.

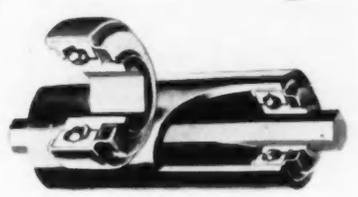
Thermospeed

A separable well tube system, known as "Thermospeed" has a temperature-sensitive bulb which is said to be 6 times faster than the average system



now in service and almost twice as fast as separable well thermocouples. In such a system where the bulb must be protected from corrosion or erosion, the extra thickness of metal and the dead air space between the well and the bulb have always caused a substantial delay or lag in transmitting a change in temperature to the bulb. This is especially true where temperatures fluctuate rapidly or on applications having small time lags and heat capacities. It is said that a dial or recording thermometer so equipped with this "Thermospeed" will be extremely accurate in indicating every temperature variation. Taylor Instrument Companies, Rochester, N. Y.

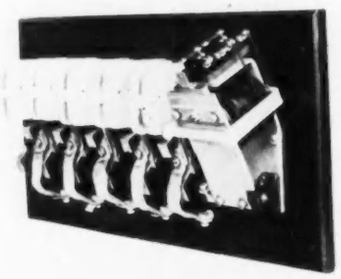
Conveyor Roller



The conveyor rollers are mounted on sealed ball bearings having hexagonally shaped holes in the inner race to accommodate a hex shaft. The rollers are made of seamless steel tubing. The bearing parts are of hardened steel having a two-piece outer race. The shell is projected over the end of the bearing to house the sealing element. Rollers of 1 in. to $7\frac{5}{8}$ in. diameter and various lengths are available. These rollers have capacities of 50 lb. continuous load rating for 1 in. diameter roller up to 8,000 lb. for the $7\frac{5}{8}$ in. diameter roller. New rollers having hexagonally shaped holes which were not available previously are: a tapered steel roller $2\frac{1}{2}$ in. to $1\frac{5}{8}$ in. diameters, having a capacity of 150 lb.; a $2\frac{1}{2}$ in. diameter roller for heavy service with a capacity of 300 lb.; a $3\frac{3}{8}$ in. diameter roller with a capacity of 750 lb.; a $3\frac{1}{2}$ in. diameter roller with a capacity of 1,200 lb.; a 4 in. diameter roller with a capacity of 3,000 lb.; and a $4\frac{1}{2}$ in. diameter roller with a capacity of 4,000 lb. Mathews Conveyor Co., Ellwood, City, Pa.

Low-Voltage A.C. Contactor

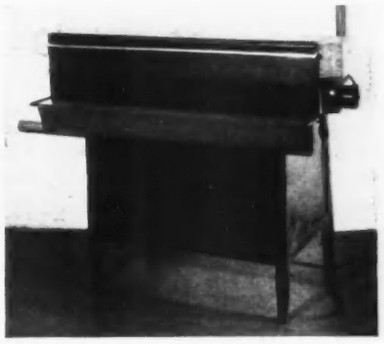
Is closed by a d.c. magnet energized through a copper-oxide rectifier and held closed by attraction of a permanently magnetized core, and the movable armature. Permanent magnet is made of the new alloy, Alnico, which retains its magnetism indefinitely, and exerts a force approximately 40 per cent greater weight for weight than other permanent magnets. A reversal of the coil exciting current by means of a pushbutton, or other pilot control device, causes the contactor to open by momentarily "bucking down" the flux of the permanent



magnet. Is said to be free from the usual transformer hum in a.c. contactors and to have no coil losses while closed. General Electric Co., Schenectady, N. Y.

Ozalid Developing Machine

For positive type prints with black, blue, or maroon lines against a white background depending on materials used. No negative step is required for reproducing prints. By the use of dry ammonia fumes as the developing agent, no liquids touch the prints at any time so that no shrinkage of paper takes place. Type 600 machines will accommodate sensitized paper or cloth of 36, 42 and 54 in. in width. The machine develops at a speed of 18 in.



per min. and weighs approximately 60 lb., is 12 in. high, 10 in. wide and 10¼ in. longer than the above sizes of paper or cloth. The feature of this size of machine is its low price and is said to be ideally suited for plants having printing requirements up to 10,000 sq.ft. per month. Ozalid Corp., 354 Fourth Ave., New York, N. Y.

MANUFACTURERS' PUBLICATIONS

Abrasion-Resisting Steel—Carnegie-Illinois Steel Corp., 434 Fifth Ave., Pittsburgh, Pa. Booklet, 11 pages, 8½x11 in. Discusses the use of alloy steel with abrasion-resisting properties for conveyors, fan blades, hoppers, screens, chutes, pipe, buckets and other applications. Chemical and physical characteristics are listed. Machining and fabricating such as shearing, punching, drilling, flame cutting, welding, forming and heat-treating are discussed. Sizes of sheets, strip, stock and plates are given.

Ball Bearings—SKF Industries, Inc., Front St. and Erie Ave., Philadelphia, Pa. Catalog No. 232, 7 pages, 8½x11 in. Self-aligning ball bearings and housings for fans and blowers. Solid and knee-type boxes are tabulated with dimensions. Types of fans, shaft diameters and other designations for 86 actual applications, adapter and cup sleeve mountings are given.

Channel Valve—Ingersoll-Rand Co., Phillipsburg, N. J. Bulletin, 8 pages, 8½x11 in. Illustrating and describing a new type of valve for air and gas compressors. The disk is slotted and recessed into which are fitted channel pieces backed by flat springs. The contruction of the valve is cleverly illustrated by six views, two of which are printed on two sheets of cellophane. These pages are so inserted to show how the channels and springs fit into the recessed slots of the valve seat and stop plate.

Corrosion-Resisting Iron — Republic Steel Corp., Massillon, Ohio. Folder on "Toncan" iron describes uses for air conditioning equipment in industrial and other applications. Both sheets and pipes made of this copper-molybdenum iron are discussed. Illustrated with a number of installations.

Conveyor Systems — Mathews Conveyor Co., Ellwood City, Pa. Book, 55 pages, 8½x11 in. "Problems Solved with Mathews Conveyor Systems." Presentation of a range and variety of typical installations of conveyor systems. The book is profusely illustrated with photographs and line diagrams of both plant and conveyor layouts. Many industries are represented for all types of manufactured products. A number of special devices are included.

Precision Needle Roller Bearings—Norma-Hoffman Bearings Co., Stamford, Conn. Folder 8½x11 in. Needle bearings for use where minimum outside dimensions, minimum weight, concentrated load capacities are requisite. Applicable to wrist pin end of connecting rods, cam rollers, rocker arms, pump shafts, gears, engines, printing machinery and miscellaneous machine purposes. Sizes 12 to 150 mm. bore available. Table of sizes and load ratings included.

Recessed Head Self - Centering Screws and Bolts — American Screw Co., Providence, R. I. Folder, 8 pages, 3x6 in. Shows the principle of the Phillips screw in which a 4-way tapered recess, which exactly fits the tapered driver, takes place of the slot in the conventional screw. Features stressed are faster driving, greater holding power, better appearance, reduced spoilage and fewer accidents. Types available are wood screws, machine screws, stove bolts and sheet metal screws. Hand and power drivers are also shown.

Rotary Pumps—Worthington Pump and Machinery Corp., Harrison, N. J. Bulletin W-475—B10, 11 and 12, 8 pages, 8½x11 in. Double-helical gear pumps with internal sleeves, external ball bearings, or internal roller bearings. Used for pumping acids, oils, dyes, syrups, glue, lacquer and paint, liquors, inks and other liquids in varying capacities. Cross-section cuts show the construction of these three types.

Rust-Proofing by the Cromodine Process — American Chemical Paint Co., Ambler, Pa. Bulletin No. 7-7, 18 pages, 8½x11 in. A comprehensive bulletin covering all the phases of this process for rust-proofing steel, cast iron, malleable iron and forged steel products. Of the 26 illustrations 12 are printed in black, silver and rust color which show clearly the results of both accelerated and actual service tests.

Silent Chain Drives—Link-Belt Co., 519 N. Holmes Ave., Indianapolis, Ind. Book, 32 pages, 8½x11 in. A list of "Silverstreak" silent chain drives obtainable from stock for ½ to 60 hp. Also complete details are tabulated for the wheels and chain constituting each drive, and horsepower and ratio tables. Simple instructions for selecting chain drives and notes on design and information of fabricated steel housings are also given.

Something New in Rubber—Something New in Plastics—Thiokol Corp., Yardville (Trenton), N. J. Folder, 6 pages, 4x8½ in. Describing a new synthetic rubber molding powder for soft molded products, resistant to oil and gasoline, lacquers, printing inks and most ordinary solvents. Parts can be molded in 3 min. at 300 deg. F. and are said to have a smooth glossy surface.

Stellite Valves—Haynes Stellite Co., Kokomo, Ind. Booklet, 8 pages, 8½x11 in. Describes the use of hard-faced valves for high temperature, high-pressure steam service. Freedom from wire-drawing, galling or seizing and corrosion is emphasized. Cobalt-chromium-tungsten alloy is used as the hard facing.

Still Tubes and High-Pressure Pipe—The Timken Steel & Tube Co., Canton, Ohio. Standard Specifications, 7 pages, 8½x11 in. Specifications for seamless hot-rolled and cold-drawn steel tubes for carrying oil at elevated temperatures and pressures in various types of oil stills, high-pressure transfer lines, steam pipe and boiler and superheater tubes. Pipes and tubes are made of chromium-silicon-molybdenum steel. Also 6 pages containing a digest of the properties of Sicromo steel with diagrams of creep strength at various temperatures.

Variable-Speed Transmission — The Oil Gear Co., Milwaukee, Wis. Bulletin 60000, 16 pages, 8½x11 in. Redesigned and simplified fluid power variable speed transmission. Photographs show interior construction. Various types are listed with dimensioned line diagrams and engineering data are included.

Vari-Pitch Texrope Sheave—Allis-Chalmers Mfg. Co., Milwaukee, Wis. Bulletin 1261, 12 pages, 8½x11 in. Variable speed device for both stationary controlled and motion controlled types; also the straight line automatic ball bearing motor base permitting complete adjustment while in operation.

BOOKS AND BULLETINS

Book of A.S.T.M. Tentative Standards

1935 Edition. 1,591 pages 6x9 in. Published by the American Society for Testing Materials, 260 South Broad St., Philadelphia, Pa. Cloth binding, $8; heavy paper covers, $7.

This annual publication of the society contains all the A.S.T.M. tentative specifications, methods of test and definitions of terms covering engineering materials and the allied testing field. The 1935 edition contains 290 tentative standards of which 75 are included for the first time. Some 65 were revised this year and are given in their latest approved form. A complete subject index has been included in the book to facilitate its use, listing items under the materials and subjects to which they apply.

Also "Proceedings of the 38th Annual Meeting." Volume 35, Part II, Technical Papers. 769 pages 6x9 in. Paper covers, $5.50; cloth covers, $6; half leather binding, $7.50. Contains symposium on spectrographic analysis with six papers and discussions. Metals are discussed in 17 papers and numerous papers and discussions on cement and concrete, ceramics, and masonry materials are also included.

A separate index to A.S.T.M. Standards and Tentative Standards is also available. This booklet has 160 pages and has paper covers.

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Turning and Boring Practice

Fred H. Colvin and *Frank A. Stanley*. 453 pages 6x9 in. 406 illustrations, 52 tables. Blue cloth-board covers. Published by McGraw-Hill Book Co., Inc., 330 West 42d St., New York, N. Y. Price $4.

Although the title implies that this book is for the shop man, the machine designer will find a wealth of material pertaining to the design and construction of various machines. These include engine lathes, turret and semi-automatic lathes, automatic screw machines and several types of boring machines. The tool and fixture designer will also find many helpful suggestions relative to these machines.

The book is divided into five sections with a total of 24 chapters. The first few sections on lathes contain 10 chapters on lathe construction, chucks, and chucking, taper, turning, thread cutting and design features of some of the newest lathes on the market. More than 150 pages are devoted to automatic screw machines. In this section cam designs are discussed, miscellaneous forming tools and their calculations, and various holding devices are illustrated. The subject of boring machines is treated in a similar manner.

In the last section of six chapters, the most recent data is given for single-point and cemented-carbide tools; speeds and machinability of metals through correct cutter design; characteristics of many of the newer metals and non-metallic materials; and a chapter on cutting oils and compounds.

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New-Method Estimators Handbook

Louis E. Almgren and Michael Wurum. 1,104 pages, 7⅜x10½ in. Cloth-board covers. Published by the New-Method Engineering Corp., 4759 Broadway, Chicago, Ill. Price $15.

First edition of a book of practical, time-saving tables arranged in simple form. Engineers, estimators and managers will find the book convenient for estimating, time study and production costs. The book is divided into nine sections, the first four relating to round, square, hexagon and flat steel stock. These sections contain tables for the linear feet of stock required per 1,000 pieces, tables of conversion to other materials and tables of weights. Similar sections and data relate to brass. One section gives production cost and time, net productions for given rates of efficiency and production time in seconds and equivalent decimal parts of a minute and hour. Several pages of standard gages are included.

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Tin Research

The nine technical publications listed are published by the International Tin Research and Development Council. Will be sent free by addressing L. J. Tavener, 149 Broadway, New York, N. Y.

"The Use of Sodium Sulphite as an Addition to Alkaline Detergents for Tinned Ware," Series A, No. 19, 12 pages.

"Examination of the Surface of Tinplate by an Optical Method," Series A, No. 21, 8 pages.

"Improvement in the Quality of Tinplate by Superimposed Electro-deposition of Tin," Series A, No. 22, 16 pages.

"The Atmospheric Corrosion and Tarnishing of Tin," Series A, No. 24, 14 pages.

"Electrodeposition of Tin Alloys from Alkaline Stannate Baths," Series A, No. 25, 9 pages.

"Equilibrium Diagrams of Binary Alloys of Tin," Series B, No. 2, 90 pages.

"Black Spots on Tin and Tinned Ware," Series D, No. 2, 27 pages.

"Solder," Bulletin No. 2, 55 illustrations, 52 pages. The last named bulletin is a comprehensive text giving the methods of soft soldering and its general uses. A number of machines are shown for soldering cans and boxes. One chapter on the use of solder in the development and manufacture of internal combustion engine radiators shows the various types of radiators and how they are assembled. A 4-page bibliography on solders and soldering contains many titles of books, research papers and reviews.

Tantalum

Fansteel Metallurgical Corp., North Chicago, Ill. Booklet, 48 pages, 6x9 in. Paper covers.

"A rare metal with amazing properties of heat and electrical transference—with unbelievable resistance to the destructive forces of acid corrosion oxidation and wear." As such is tantalum, the metallic element described in this booklet. After a short history, the uses of this pure metal are listed in the chemical and automotive industries, in the laboratory, and for the manufacture of cellulose and rayon, chlorine, and electronic tubes.

Physical properties, ductility, workability machinability, welding, hardness, and resistance to wear and erosion are given. Thermal conductivity and expansion reflectivity, radiation, and color are also discussed. Electrical, chemical, and electrochemical properties are included. In tabular form are given the technical characteristics of a number of metals, corrosion resistance tests of tantalum, wire and rod data, weight, tolerances of sheets and information on tubing. All the subjects are indexed.

V-Belt Drive Test

Air Commerce Bulletin, Vol. 7, No. 3, "Report on a 300-hr. test of an Automobile Engine with a Belt Drive Proposed for Airplane Use." Obtainable from U. S. Department of Commerce, Washington, D. C., or the Manhattan Rubber Mfg. Division of Raybestos-Manhattan, Inc., Passaic, N. J.

A 300-hr. government-supervised run in which an airplane propeller was driven from an automobile engine by six stock Condor V-Belts of the same construction as those made for commercial drives. The report states that the drive was 50 per cent underbelted, and showed only 5 per cent loss of strength. The stretch was 5 per cent under a 900-lb. load. Slippage from 3,200 r.p.m. to "wide open" averaged 3 per cent. It is also stated that the belts showed "remarkably good wear" after the test. Included in the report is a description of the construction of the belts used for this test.

Durez News

General Plastics Inc., North Tonawanda, N. Y. House organ, 8½x11 in.

Announcement of a monthly publication which tells of and shows applications of molded Durez powders in product design. Advice is given on the proper type of molding powder to be used and suggestions of molding sources. The booklet illustrates photographically various applications in a number of industries.

Double Helical Gear Calculations

EMIL R. GASSER
Engineer, Farrel-Birmingham Company

THE example shows the calculations for the design of a pair of double helical gears when the wearing quality rather than the strength of tooth is to be considered. The data given includes the speed of both shafts, horsepower, overload factor, gear and pinion materials and tooth pressure angle.

Problem—Design a pair of double helical gears to transmit 75 hp., the speed of the high-speed shaft is 870 r.p.m., and that of the low-speed shaft 110 r.p.m. The gears are to run eight hours a day and must withstand occasional overloads of 100 per cent for short intervals. The gear is to be a steel casting and the pinion a heat-treated steel forging.

It is assumed the gears are to be cut on a Sykes generator equipped to cut 20 deg. pressure angle and a 30 deg. helix angle. The tooth addendum is to be $0.80/DP$. The outside diameter of the gear will be pitch diameter $+0.80/DP$ and the outside diameter of the pinion pitch diameter $+2.40/DP$.

Preliminary calculation of pitch diameter of pinion, if no center distance is given:

$$PDp = 7.5\sqrt[3]{\frac{hp.}{Cr \text{ r.p.m. of pinion}}}$$

PDp = Pitch diameter of pinion

$$Cr = \text{Ratio factor} = \frac{R}{R+1}$$

$$R = \text{Ratio of } \frac{\text{r.p.m. of pinion}}{\text{r.p.m. of gear}}$$

$$PDp = 7.5\sqrt[3]{\frac{75 \times 8.9}{7.9 \times 870}} = 3.450 \text{ in.}$$

When calculating the pitch, it should not be finer than the following:

$$DP = \frac{0.04 \times F \times PDp \times r.p.m.}{hp.}$$

F = Face width. Is assumed to be two times the pitch diameter of the pinion. Preliminary face width = 6.9 in.

Minimum DP = Diametral pitch in plane of rotation

$$\text{Minimum } DP = \frac{0.04 \times 6.9 \times 3.45 \times 870}{75} = 11DP$$

To allow for overloads and possible shocks, this value of DP is cut to approximately one-half, giving a pitch of from 5 to 6 DP. Choosing 6 DP and 21 teeth for the pinion, the pitch diameter is:

$$PDp = \frac{21}{6} = 3.500 \text{ in.}$$

The number of teeth in the gear will then be:

$$\frac{21 \times 870}{110} = 166 \text{ teeth}$$

The center distance C of the pair of gears will be:

$$\frac{21 + 166}{2 \times 6} = 15.583 \text{ in.}$$

If this preliminary center distance is satisfactory for the installation, it can be taken as final.

Corrected Face Width

The correct width of tooth face is now computed from the equation:

$$F = \frac{P}{Cm\ Cr\ Cc\ Cq\ Cv\ Ci\ PDp}$$

P = Tangential tooth load in the plane of rotation

$$P = \frac{126{,}000 \times \text{hp.}}{r.p.m. \times PDp}$$

$$Cm = \text{Material factor} = \frac{\overline{Ss}^2}{958^2}$$

$$Ss = \text{Allowable shear stress} = \frac{Sy}{\sqrt{f}} \text{ or } Se$$

f = Factor of safety
Sy = Shear yield point
Se = Endurance limit in reverse shear
Cr = Ratio factor
Cc = Curvature factor = $sine\ \phi\ cos\ \phi$
ϕ = Pressure angle in plane of rotation

$$Cv = \text{Velocity factor} = \frac{152}{152 + \sqrt{PDp\ r.p.m.}}$$

$$Ci = \text{Face factor} = \frac{49 - F}{72}$$

For gear faces 2 in. and less, $Ci = 0.65$
For gear faces 18 in. and greater $Ci = 0.43$

$$Cq = \frac{CL\ min.}{F} \text{ or approximately } Cq = \frac{0.9 \times LA}{NBP}$$

$CL\ min.$ = Minimum length of contact
LA = Length of line of action =
$\frac{1}{2}\sqrt{ODp^2 - PDp^2 \cos^2 \phi} +$
$\frac{1}{2}\sqrt{ODg^2 - PDg^2 \cos^2 \phi} - C \sin \phi$

NBP = Normal base pitch = $CP \cos \phi \cos \beta$

$= \dfrac{\pi}{DP} \cos \phi \cos \beta$

β = Base helix angle

$Tan\ \beta = Tan\ \alpha \cos \phi$

α = Helix angle at pitch circle

ODp = Outside diameter of pinion

ODg = Outside diameter of gear

PDg = Pitch diameter of gear

Example of Calculations

The various factors are now calculated. It is assumed that the material in the gear has a shear yield point Sy of 42,000 lb. per sq.in. and an endurance limit in shear Se of 32,000 lb. per sq.in. The safe stress Ss with a factor of safety of 2 is:

$$Ss = \frac{42{,}000}{\sqrt{2}} = 29{,}700 \text{ lb. per sq.in.}$$

Since this value of Ss is less than Se, it should be used to obtain the material factor Cm. The pinion has a shear yield point Sy of 55,000 lb. per sq.in., and an endurance limit Se of 34,000 lb. per sq.in., the safe stress Ss with a factor of safety of $2\frac{1}{2}$ for the pinion is:

$$Ss = \frac{55{,}000}{\sqrt{2.5}} = 34{,}800 \text{ lb. per sq.in.}$$

The gear is weaker than the pinion, therefore Ss is taken as 29,700 lb. per sq.in. Cm then is:

$$Cm = \left(\frac{29{,}700}{958}\right)^2 = 961$$

$$Cr = \frac{7.9}{8.9} = 0.888$$

$$Cc = Sin\ 20^\circ \cos 20^\circ = 0.321$$

$$Cv = \frac{152}{152 + \sqrt{3.5 \times 870}} = 0.735$$

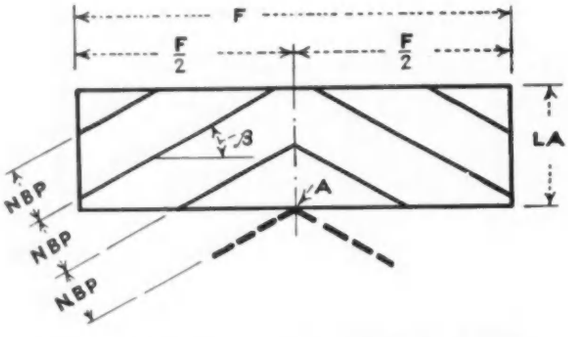

Fig. 1—The amount of tooth contact can be determined by scaling diagram drawn from dimensions of face width F, length of line of action LA, normal base pitch NBP, and tooth helix angle

$$Ci = \frac{49 - 7}{72} = 0.584 \text{ assuming the face to be 2 } PDp$$

The outside diameter of the 21-tooth pinion will be $PDp + (2.40/DP)$ or $(21 + 2.40)/6$. The outside diameter of the 166-tooth gear will be $PDg + (0.8/DP)$ or $(166 + 0.8)/6$.

$PDp = 21/6 = 3.500$ in. $\quad PDg = 166/6 = 27.667$ in.

$ODp = 3.500 + 0.400 = 3.900$ in. or $23.4/6 = 3.900$ in.

$ODg = 27.667 + 0.133 = 27.800$ in. or $166.8/6 = 27.800$ in.

Then LA, the length of action is:

$$LA = \tfrac{1}{2}\sqrt{3.9^2 - 3.5^2 \cos^2 20^\circ} + \tfrac{1}{2}\sqrt{27.8^2 - 27.666^2 \cos^2 20^\circ} - 15.583 \sin 20^\circ$$

$LA = 0.64$ in.

$Tan\ \beta = Tan\ 30^\circ \cos 20^\circ = 0.5425$

$\beta = 28$ deg. 29 min.

$$NBP = \frac{3.1416}{6} \cos 20^\circ \cos 28^\circ 29' = 0.432 \text{ in.}$$

$$Cq = \frac{0.9 \times 0.64}{0.432} = 1.33$$

$$P = \frac{126{,}000 \times 75}{870 \times 3.5} = 3{,}120 \text{ lb.}$$

The face width F is:

$$\frac{1}{Cm\ Cr\ Cc} = \frac{1}{961 \times 0.888 \times 0.321} = \frac{1}{274}$$

$$\frac{1}{Cq\ Cv\ Ci\ PDp} = \frac{1}{1.33 \times 0.735 \times 0.584 \times 3.5} = \frac{1}{1.996}$$

$$F = \frac{3120}{274 \times 1.996} = 5.70 \text{ in.}$$

This value of 5.70 in. for the width of gear face is less than twice the pitch diameter of the pinion, which is preferable. A final check is now made, taking the face width as $5\frac{3}{4}$ in. and at the same time revising Ci, the face factor.

$Cm\ Cr\ Cc = 961 \times 0.888 \times 0.321 = 274$

$Cq\ Cv\ Ci = 1.33 \times 0.735 \times 0.601 = 0.586$

$PDp^2\ r.p.m.\ F = 3.5^2 \times 870 \times 5.75 = 61{,}200$

$$Hp. = \frac{274 \times 0.586 \times 61{,}200}{126{,}000} = 78.2\ hp.$$

The factor $Cq = CL\ min/F$ can be determined graphically when the exact value is used in place of $Cq = 0.9 \times LA/NBP$. The following method is very simple:

The length of line of action LA, the normal base pitch NBP, and the base helix angle, are calculated from formulas given. The minimum length of contact is obtained when a new tooth is just about to engage.

By drawing a rectangle with the sides F and LA, and starting at point A, the lengths of the lines of contact of each tooth can be scaled, and the total is $CL\ min.$, the minimum length of contact.

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